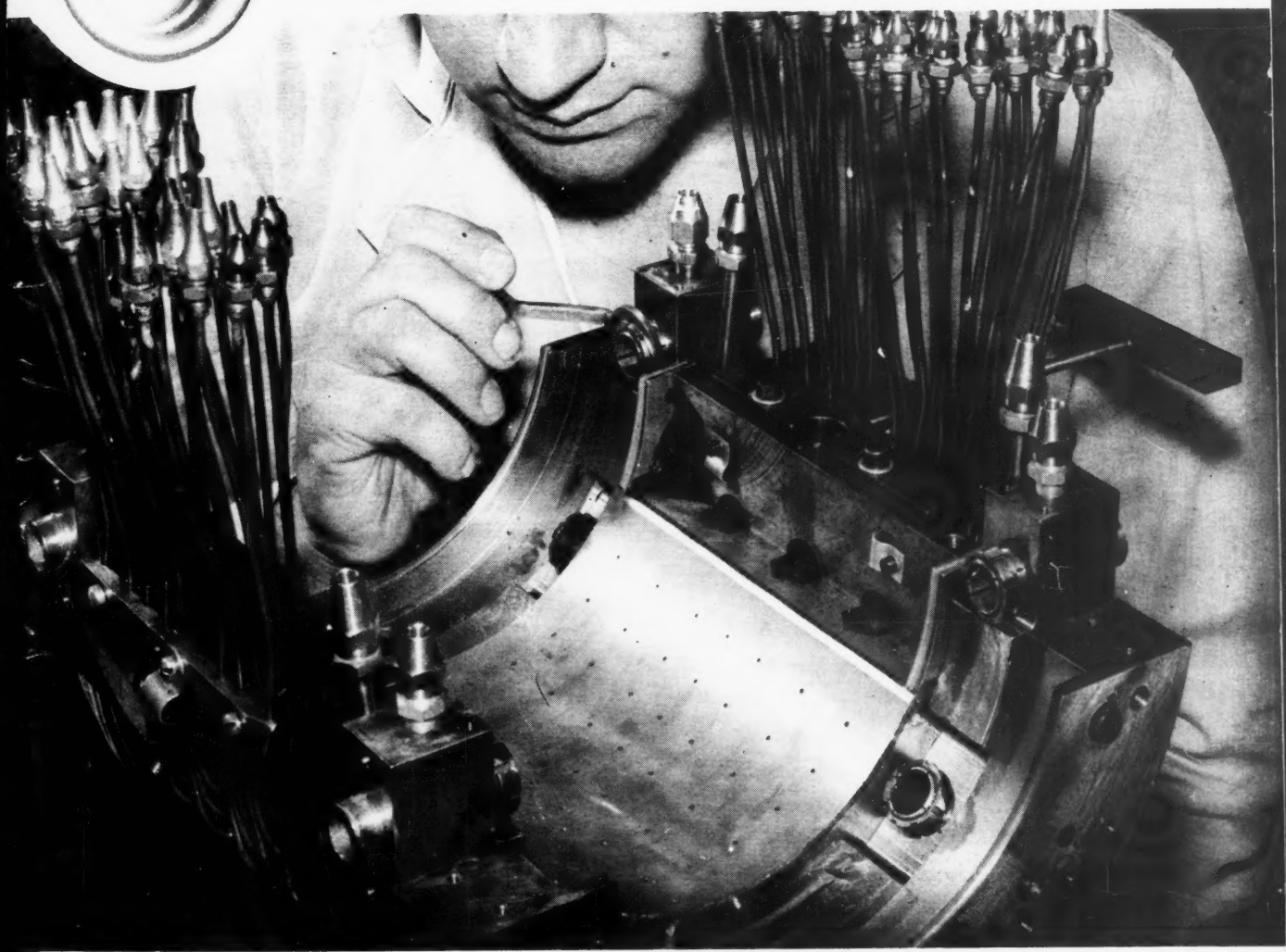
1954 Subject & Author Indexes (p. 359)
November-December, 1954
Vol. 10, No. 6

Lubrication ENGINEERING



Bearing Materials (p. 313)

JOURNAL OF
THE AMERICAN SOCIETY OF LUBRICATION ENGINEERS

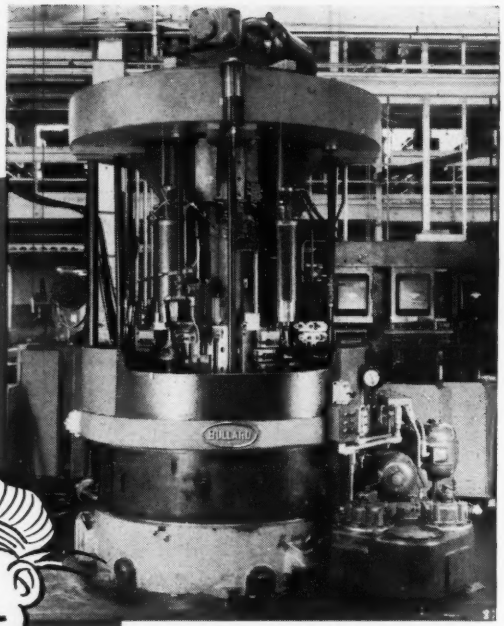


Photo courtesy Bullard Machine Tool Co.

Photo courtesy Cross Co.

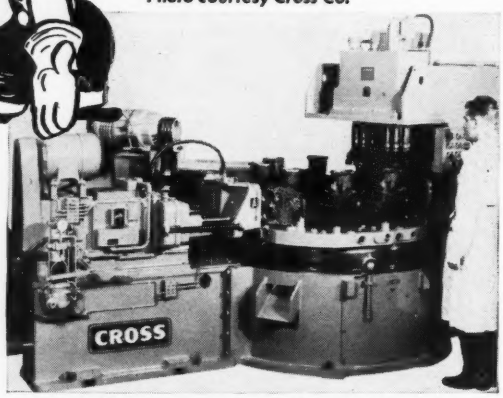
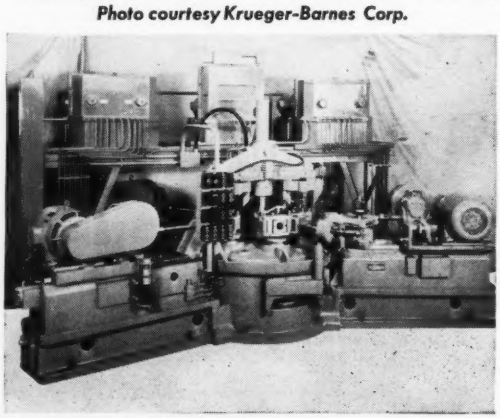


Photo courtesy Krueger-Barnes Corp.



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Lubrication ENGINEERING

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(Cover Picture courtesy of Westinghouse Research Laboratories)

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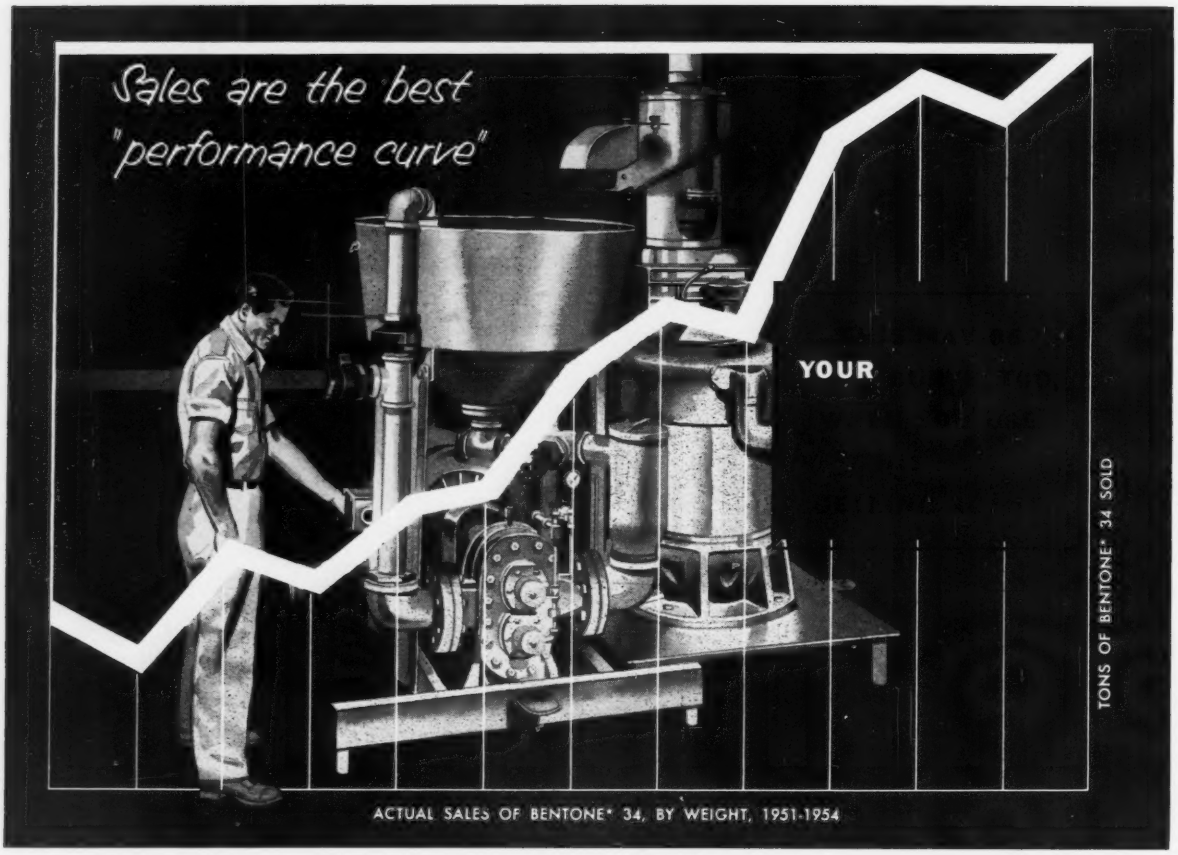

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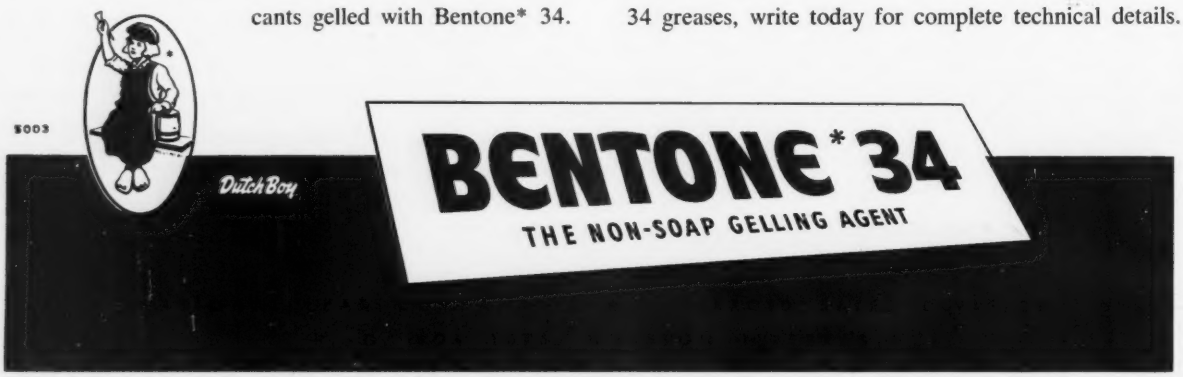
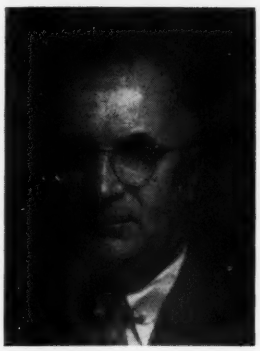


Meet the Authors



R. A. Baudry
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O. W. Wuerz
Cab Service & Parts Corp.
(p. 319)

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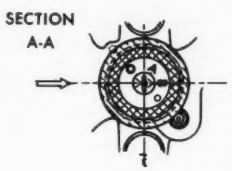
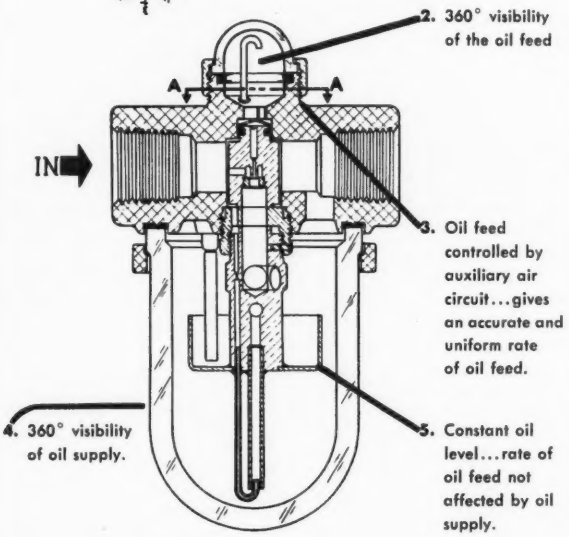


1. VARIABLE VANE VENTURI View showing Index Marks "A" and "B" for Position of Rotatable Vane Type Venturi Plug.



2. 360° visibility of the oil feed

3. Oil feed controlled by auxiliary air circuit...gives an accurate and uniform rate of oil feed.

4. 360° visibility of oil supply.

5. Constant oil level...rate of oil feed not affected by oil supply.

6. Ability to produce a very fine oil fog and deliver it over long distances, through complex piping systems, and obtain equal distribution to multiple lubrication points.

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- ***each can handle jobs now done by several sizes of lubricators***

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PERMAGEL

Lube Lines

(**Editor's Note:** Information for this feature has been coordinated by **Erwin Landau,** Asiatic Petroleum Corp., 655 Fifth Ave., New York 22, N. Y. All readers who have any developments or personnel changes they would like to publicize, should send the details to Mr. Landau at the above address.)

'LUBRICATION ENGINEERING' PASSES 1st DECADE. Lubrication Engineering is celebrating the completion of ten continuous years of publication with this issue. Of interest to all is a Decennial Subject & Author Index that will be included as a supplement to the January-February '55 issue.

WHILE REMINISCING. In October the Oil Industry Information Committee held its annual public relations week and two tidbits of information which might have passed unnoticed are: More than half of the oil wells in the United States are still producing although some are more than 50 years old . . . Thirty years ago, the oil industry employed but 200 research technologists. Today, oil companies in the U. S. employ over 15,000 in research activities . . .

FINANCIAL RISKS IN RESEARCH PROJECTS. In a recent speech, Dr. Gustav Egloff, research director of Universal Oil Products Co., Des Plaines, Illinois, stated that only one research project in 20 reaches a financially successful conclusion. He cautioned that a smaller company considering a new research department should bear in mind that a considerable amount of money may be spent and much time may elapse before the venture is financially successful. One of the world's largest companies estimates that it spends four times as much on unsuccessful research projects as it does on successful ones.

NAVY INQUIRY. Here's a chance to tell the Navy how they ought to inspect lubricants. They have asked for suggestions on improving the present procedure. The Navy wants to know how oil companies guard product quality from refinery to customer. Send all thoughts to: Bureau of Aeronautics, Navy Department, Washington 25, D. C.

MOTOR VEHICLE REGISTRATIONS ON INCREASE. The Bureau of Public Roads has reported that motor vehicle registration in the United States should reach 58,129,000 for 1954, including 48,087,000 passenger cars, which would be an increase of about 3.5% over 1953.

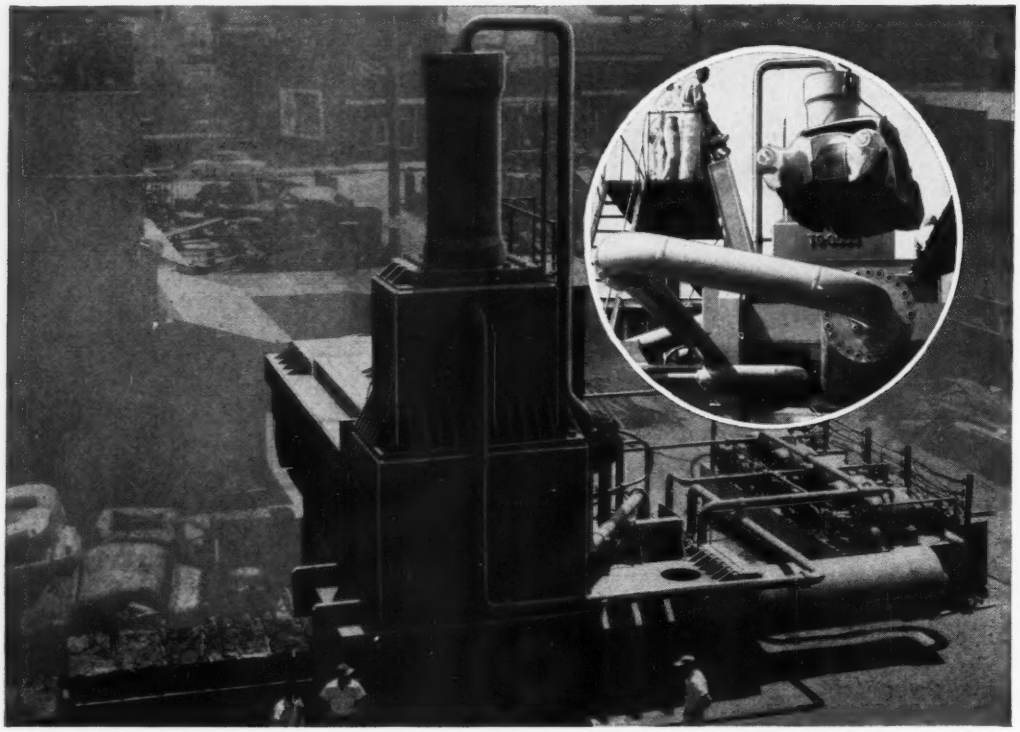
CREDIT LINE. Kurt H. Straus of The Texas Company assisted in the editorial revision of several of the technical papers appearing in this issue.

FOR THE CURIOUS: Did anyone ever question how his state rates in consumption of lubricants? Shown below are estimated millons of gallons of automotive and industrial lubricants consumed by states, as published in the API Tax Economics Bulletin, Vol. 19, No. 4, July-August, 1954.

	Automotive	Industrial
Alabama	15.5	15.5
Arizona	7.2	1.5
Arkansas	10.1	2.8
California	91.4	40.4
Colorado	12.1	3.2
Connecticut	13.1	15.9
Delaware	2.6	2.6
District of Columbia	4.8	0.7
Florida	24.0	5.0
Georgia	21.1	12.0
Idaho	5.1	1.2
Illinois	58.2	57.2
Indiana	31.4	34.5
Iowa	23.0	6.8
Kansas	18.8	5.8
Kentucky	14.8	6.4
Louisiana	14.7	10.8
Maine	5.7	7.7
Maryland	14.5	13.5
Massachusetts	23.4	27.6
Michigan	47.8	55.4
Minnesota	22.6	10.3
Mississippi	11.3	3.3
Missouri	30.8	14.0
Montana	5.9	1.5
Nebraska	11.5	2.3
Nevada	2.4	0.9
New Hampshire	3.2	3.4
New Jersey	33.9	40.0
New Mexico	6.6	0.5
New York	66.7	70.0
North Carolina	24.9	19.1
North Dakota	6.4	0.2
Ohio	56.1	72.9
Oklahoma	18.0	3.8
Oregon	13.2	7.9
Pennsylvania	55.9	80.2
South Carolina	13.0	11.3
South Dakota	6.7	0.5
Tennessee	18.1	12.3
Texas	89.3	24.0
Utah	5.6	2.4
Vermont	2.5	1.7
Virginia	22.2	12.4
Washington	17.4	12.6
West Virginia	9.1	10.2
Wisconsin	24.2	22.4
Wyoming	3.6	0.4

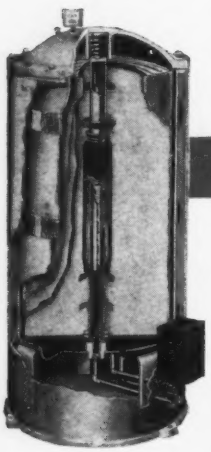
NEW ASLE PUBLICATION: Fundamentals of Friction & Lubrication in Engineering (Proceedings of 1st ASLE National Symposium) has just been published by ASLE and is available to members at $3.00 per copy, to non-members at $3.50 per copy. See page 360 for a summary of the contents.

world's largest scrap-baling press gets HONAN-CRANE Oil Purification!



An outstanding example of industry's modern, push-button miracles is the new TG-3000, gigantic scrapmetal baling press specially built by the Harris Foundry & Machine Company for the Chicago Auto Wrecking Corporation.

Weighing 675,000 pounds, the huge, 800-horsepower press is capable of baling up to 75 tons per hour. It actually can compress two or more complete autos simultaneously into neat packages measuring about 24″ x 24″ x 60″ each.



HONAN-CRANE OIL PURIFIER PROTECTS 3700 GALLON HYDRAULIC SYSTEM

The TG-3000 requires a hydraulic system containing 3700 gallons of high grade hydraulic oil. Push-button controls connected to the hydraulic circuit make the press manual, semi-automatic or fully automatic in operation.

To insure against costly breakdowns in this complex system, the Harris Company will equip it with a Honan-Crane Bulk-Type Oil Purifier as shown at left. Honan-Crane purification protects vital hydraulic mechanisms, extends safe-use life of oil, insures maximum operating efficiency and economy.

Give your plant the benefit of Honan-Crane's 30 years' filtration experience. There's a Honan-Crane Purifier for complete removal of either solid or soluble contaminants from hydraulic, lubricating, metal-working, insulating and other oils.

Write for engineering bulletins covering your specific problem

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"Scrap and Pig for the Furnaces." Agnes Potter Lowrie depicts a variety of objects which will emerge Phoenix-like from the flames of the open hearth to new life as virgin steel: buck-toothed gears, a section of track, machine shop shavings, old tubing, stampings and, bulking equally large, rusty red pig. Limited edition of 15 x 18 color prints. Copy on request.

The Fine Art of Steel Making #IV

In sharp contrast to many industries, the basic ingredients of steel making have changed little, if any, over the years. How then has steel been able to spearhead America's industrial growth? How has it been able to produce more steel, better quality steel year by year?

Because steel men have worked constantly to improve the *mechanics* of steel making, as well as the metallurgy. And because steel's suppliers have parallelled every such improvement with an equal advance of their own.

Consider the materials which smooth the process of steel making — Ironsides Gear Shield lubricants. Heavy Gear Shield, once hand-paddled onto exposed gears, is now formulated with a solvent for quick spray application.

For encased gears, Gear Shield was produced in liquid form for pouring, pumping or timed-jet application. Later Ironsides was among the leaders in developing extreme pressure lubricants.

Today, so widespread is the acceptance of Ironsides lubricants that other makers have attempted to refer to their own products as "gear shield", even though Gear Shield is an Ironsides trade mark.

Ironsides is not mass production. We are the "Custom Tailors" of lubrication. Our special position is due in large part to our flexibility. We can — and do — formulate for individual applications and supply these formulas in any quantity from pails to tank cars.

We like tough problems; we've solved a lot of them. For example, Palmoshield, replacement for palm oil and most important advance in lubrication since World War II. If you have a problem, we'd like to help lick it. A letter will summon one of our research engineers. The Ironsides Co., Columbus 16, Ohio.

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- AIR COMPRESSORS
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- TRANSFORMERS
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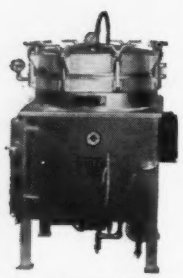
- WIRE DRAWING MACHINES
- HYDRAULIC EQUIPMENT
- METAL ROLLING MILLS
- PAPER MAKING MACHINERY



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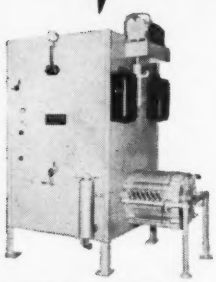
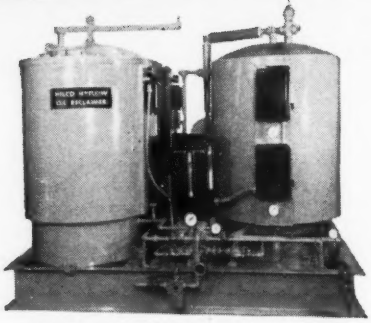


PURIFIER-RE-REFINER

For oil purification in batches of from 6 to 100 gallons. Removes all solid and volatile contaminants.

FILTER

Furnished in capacities from 0.1 to 750 gpm. Various cartridges available for mineral and inhibited detergent oils.



HIGH CAPACITY RECLAIMER

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Hilco units are available for continuous or batch operation. You can keep your lubricating and industrial oils clean and recover large quantities of oil at low cost.

Wherever Oil is used it becomes contaminated — must be discarded or conditioned for further use.

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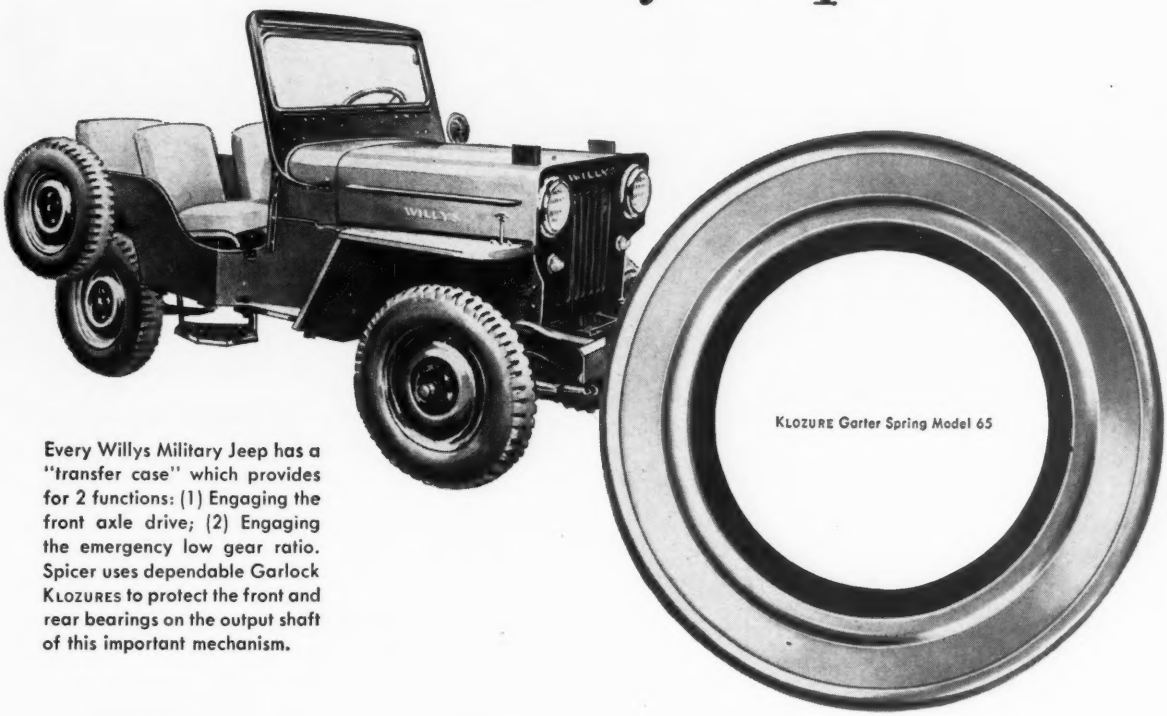
For *positive* bearing protection on Military Jeeps-



Every Willys Military Jeep has a "transfer case" which provides for 2 functions: (1) Engaging the front axle drive; (2) Engaging the emergency low gear ratio. Spicer uses dependable Garlock KLOZURES to protect the front and rear bearings on the output shaft of this important mechanism.

Spicer uses KLOZURE* Oil Seals

The Spicer "transfer case" in a Military Jeep must be able to function smoothly under the most adverse operating conditions. Thus, the bearings on the output shaft (which operates at speeds up to 4,000 R.P.M.) need *positive* protection against dust, mud, and water; the bearing lubricant must be sealed in. This job calls for a superior oil seal. Spicer engineers rely on KLOZURE garter-spring Model 65.

Take a tip from Spicer, who for 50 years has been manufacturing precision parts for the automotive industry. Standardize on KLOZURE Oil Seals—the best bearing protection money can buy.

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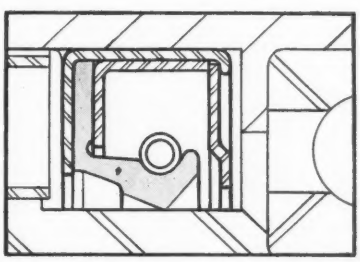

Installation of Model 65 KLOZURE Oil Seal on the output shaft bearing of the Spicer "transfer case."

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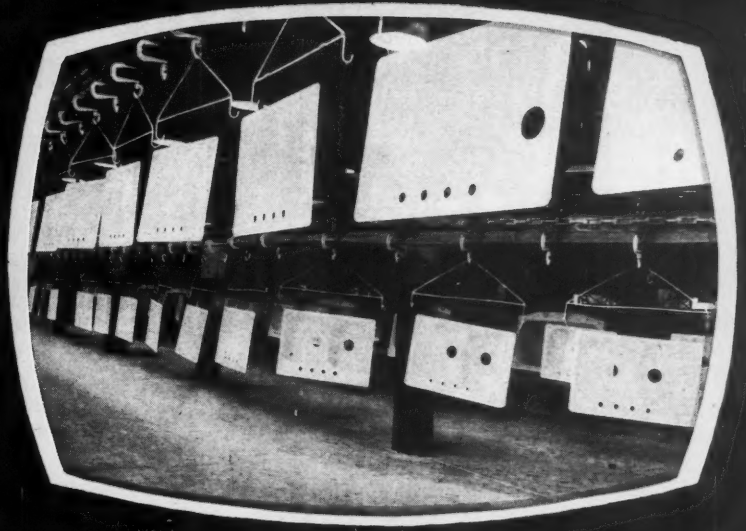

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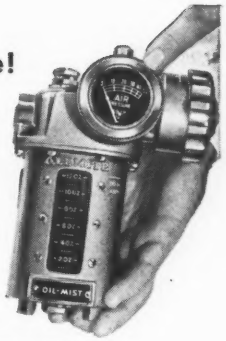

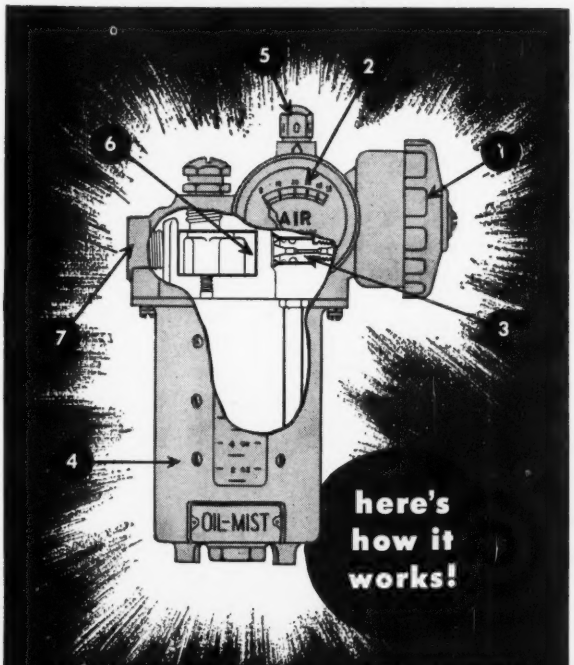

This Oil-Mist Lubricator atomizes oil into microscopic particles which are carried in the air stream and distributed through tubing to the bearings. Compressed air enters and passes through the air regulator (1) and air gauge (2). As this air passes through venturi (3) it draws oil from the reservoir (4). Oil flow is set by knob (5). The mixture of oil and air from the venturi is thrust against the baffle (6). Only the most minute, lighter-than-air particles are blown through the outlet (7) into the delivery line to bearings.

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A complete range of models and multiple unit models, in capacities of 12 ounces and one gallon, assure you a perfect choice for any application!

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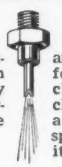
allow the use of Oil-Mist on any machine!

Oil-Mist Fittings bring the most efficient lubrication in the world to any anti-friction bearing—roller, needle or ball.



Spray Fittings are recommended for open and enclosed gears and chains. Allow for a concentrated spray of oil where it is needed.

Condensing Fittings apply oil in liquid form to plain bearings, slides, ways, vees, cams and rollers.

LUBRICATOR SPECIFICATIONS:

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- Material—die cast aluminum body with nylon plastic window.
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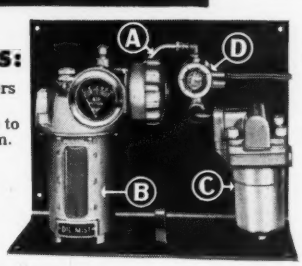
- Solenoid Control (D) starts system automatically when machine starts—foolproof.



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ASLE Annual Meeting & Exhibit

TENTATIVE PROGRAM

April 13,14 & 15, 1955 **Hotel Sherman, Chicago, Ill.**

Sixteen technical sessions, a five-session educational course, three panel discussions, a symposium, and a plant tour have been tentatively scheduled as the Program for the 10th Annual Meeting & Lubrication Exhibit of the American Society of Lubrication Engineers to be held in Chicago on April 13, 14 and 15, 1955.

The Panels will include: (1) "Controls for Extending the Useful Life of Coolants While in Use," (2) "Industrial Dermatitis," and (3) "Organization of a Plant Lubrication Program," with D. M. Cleaveland & E. L. H. Bastian; J. W. Hopkinson; and K. S. Smiley, as organizing chairmen, respectively.

The Symposium, with A. F. Brewer as organizing chairman, will cover "Plant Safety as Related to Lubrication."

Technical Sessions will be devoted to: (1) Film-Type Bearings, (2) General Industry, (3) Greases, (4) Hydraulic Fluids, (5) Lubricant Dispensing Equipment & Application, (6) Measuring Properties of Lubricants, (7) Metal Working, (8) Modern Concepts of Wear, (9) Present Thinking in Gear Lubrication, (10) Rolling Contact Bearings, and (11) Timely Studies in Friction. Tentative papers include:

Adhesive Wear of Metals, by C. D. Strang, Jr.

Atoms Trace the Wear, by H. Jackson & W. Miller

Centralized Grease Lubrication Equipment, by L. T. McDonald

Control of the Quality & Life of Grinding Coolants During Use, by O. L. Maag

Control of Quality of Water Soluble Machining Coolants During Use, by J. A. Prevel

Corrosion Inhibited Greases, by G. H. Link

Critical Survey of Industrial Lubrication Engineering in Great Britain and the U.S.A., by G. D. Jordan

Economic Factors of Centralized Lubrication, by R. Lohbauer

Effect of Additives in Mineral Oil on Rate of Fretting Wear, by D. Godfrey

Effect of Pressure & Molecular Weight Upon the Viscosity of Polybutens, The, by L. B. Sargent

Effect of Sliding Velocity & Temperature on Wear & Friction of Several Materials, The, by R. L. Johnson, M. A. Swikert & E. E. Bisson

Evaluation of Lubricating Grease Compatibility, by A. L. McClellan & S. R. Calish

Factors Influencing Friction & Wear with Solid Lubricants, by M. B. Peterson & R. L. Johnson

Frictional Characteristics of Plastics, by W. C. Milz & L. P. Sargent, Jr.

Future Trends in Sealed Ball Bearings, by L. D. Cobb

Gear Lubrication in the Steel Industry, by J. P. Critchlow

Heat Treating & Quenching Oils, by R. Weir

High Speed Friction & Wear Machine, A, by H. G. Clarke, Jr. and W. W. Shugarts, Jr.

Influence of Moisture on the Friction and Surface Damage of Clean Metals, The, by R. O. Daniels and A. C. West

Laboratory Appraisal of Drawing Compounds, by W. J. Wojtowicz

Low Temperature Operation of Aircraft Accessories, by E. A. Bancak and R. S. Barnett

Lubricants Dispensing Equipment in Metal Working Machinery Operations, by M. A. Cavanaugh

Lubrication of Rolling Contact Bearings at Elevated Temperatures with Solid, Liquid and Gaseous Lubricants, by Z. N. Nemeth & W. J. Anderson

Maintenance of Quality in Aluminum Sheet Rolling Oils, by J. O. McLean

Mass and Energy Balance for the Wear Process, A, by P. F. Chenea

Modern Water Based Cutting Fluids, by A. Manteuffel

Nature and Performance of Some Inorganic Base Greases, by W. H. Peterson and A. Bondi

New Bearing Metal Theory in Which Lubrication is an Important Factor, A, by B. Lunn

New Measurements of Attitude and Eccentricity in Journal Bearings, by B. Sternlicht and D. D. Fuller

Non-Flammable Hydraulic Fluids (two papers), (1) by W. H. Millett, and (2) by C. F. Schmidt

Progress Report on Silicone Lubricants, A, by H. F. Lamoreaux

Pumpability of Aircraft Turbine Lubricants at Low Temperatures, by J. Bridges and E. S. Starkman

Quantitative Method of the Measurement of Sulfur Activity of Lubricants, A, by R. S. Barnes

Radioactive Wear Studies, by A. Hundere

Safety Aspects of Centralized Lubrication, The, by T. G. Carico

Some Unusual Conditions Encountered in the Lubrication of Rolling Contact Bearings, by T. W. Morrison

Study of the Sliding Process of Metals, A, by L. F. Coffin, Jr.

Survey of Wear Mechanisms, by J. T. Burwell, Jr.

Tool Weight Loss for Cutting Fluid Evaluation, by J. P. G. Beiswanger, J. P. Copes & R. L. Mayhew

Use of Glass as a Lubricant in Extruding Steel, The, by J. Strauss

Wear of Iron in Sliding Contact with Various Elemental Metals, The, by C. L. Goodzeit, R. P. Hunnicutt and A. E. Roach

Wear Studies with Radioactive Gears, by V. N. Borsoff

Worm Gear — Operation and Lubrication, The, by S. D. Craine and R. B. Moir

X-Ray Methods Applied by Automotive & Petroleum Research, by F. W. Lamb

BEARING MATERIALS†

by A. F. Brewer*

Both metallic and non-metallic bearing materials available to modern industry are reviewed and their advantages, limitations, and general lubrication requirements are discussed.

Progress in the development of bearing materials has been most interesting since Isaac Babbitt announced his formula for Babbitt metal over a hundred years ago. As machine speeds were increased, and as temperature and pressure ranges were widened, it was only natural for the designing engineer to look to metallurgy for alloys which would function most dependably under the expected operating conditions. The cooperative research which resulted led first to the admiralty bronzes and later to the variety of high tin-base and high lead-base metals alloyed with varying amounts of antimony or copper according to the service involved. These were augmented some 20 years ago by copper-lead and by the cadmium alloys of nickel and silver which permitted higher unit loadings. Copper-lead took its place as one of the major bearing metals when World War II cut off the main source of tin—the Dutch East Indies.

Quite naturally non-metallic materials also must be considered in any general discussion of sleeve-type bearings. From wood—the earliest known—to the laminated synthetics and nylon, the transition makes an interesting story. The approximate sequence in which the present-day bearing materials came into accepted usage is as follows: (1) Wood, (2) Bronze, (3) Babbitt Metal, (4) Copper-Lead, (5) Porous Metals, (6) Rubber, (7) Aluminum Alloys, (8) Silver, (9) Graphited Metals, (10) Phenolic-Base Materials, (11) Nylon.

All bearing materials have their own specific characteristics, certain of which make them particularly suited for definite applications. However, all must possess two basic features: low coefficient of friction and ability to withstand shaft load and sudden impact. Pliability in the material assures conformability. An essential requirement in any bearing material, exclusive of design, is that it should be softer than the shaft or journal so that in case of uneven set-up, too little clearance, overheating or presence of abrasive foreign materials, the bearing will score, wear, wipe, or flow before the steel shafting becomes scored or worn. It is cheaper to renew bearings than shafting.

WOOD. Wood as a bearing or axle supporting medium probably came into usage when the wheel was invented many centuries ago. Very likely the first wheel hubs used on the chariots of antiquity were mounted on wooden axles. It can be assumed that the value of lubrication was discovered soon after, when animal fats were found to keep the wheels from squeaking or even sticking to the axles under fast moving (horse running) conditions.

About the time of Rameses II, 1300 B.C., iron bearings came into usage on the chariots of that period. Wood, however, continued to be used as crude thrust bearings and as solid wood wheels on carts and water wheels. As the machine age progressed, overhead shafting, in turn, was run through holes in wooden blocks. Until this very day hard woods such as maple are used for special bearing applications on certain textile, food processing, and dairy machinery, the wood being impregnated with straight mineral oil of medium viscosity 65-70 centistokes (300 SSU at 100 F.) to give self lubrication. In marine service, however, the lignum vitae wood stern tube bearings are designed so that sea water can circulate through the bearing clearance and thus provide ample lubrication.

The use of oil-impregnated wood bearings of maple, or some similar hard wood such as oak, has been a development which proved economical for applications where bearing metals and re-lubrication might present a problem and where ample clearance is practicable. Oil-impregnated wood bearings can be replaced at low cost when renewal of the lubricated bearing surface is considered necessary. Temperature control in such a bearing is important because of the low thermal conductivity of wood. Operating temperatures in excess of 150 to 200 F. can cause the surface to char and affect the porosity and self-oiling ability.

BRONZE — THE COPPER-BASE METAL. When machinery began to be made of iron and some combinations of non-ferrous metals, the need for metallic bearing materials was indicated. There was a gradual increase in speed and load as the machine age progressed into the nineteenth century. These conditions prompted the further usage of bronze as a bearing metal, from the status of one of

*(Consulting Lubrication Engineer) 314 Second Ave., Avon-by-the-Sea, N. J.

†This article constitutes one of the chapters in the forthcoming book, **Basic Lubrication Practice**, by Mr. Brewer, to be published by The Reinhold Publishing Corp. early in 1955.

the oldest known bearing materials to that of the most useful. Bronze enjoyed this preference until Babbitt metal was invented in 1839.

The major constituent of any bronze is copper. The earlier metals of this type were comparable to the well known "Admiralty Bronze" or "Gun Metal" which contained upward of 90 percent copper, with tin and zinc making up the rest of the alloy. Table I lists the modern copper-base bronzes with their identity, average composition, and relation to lubrication.

Bronzes as a whole are tough, durable materials with good anti-friction qualities resistant to wear and deformation when adequately lubricated. The rate of wear under conditions of boundary lubrication is decreased by increase in the tin or lead content. The bronzes are among the hardest of the present-day bearing metals, with good compressive strength and high melting point. They may tend to run hotter than the so-called "white metals" (Babbitts), however, so they are used mainly where high melting point and strength are desired. Due to their hardness they do not seat as readily to the rotating shaft, as do the softer lead and tin base Babbitts; consequently the shaft may ride on a number of high points. Reduction of the effective bearing area may cause overheating.

BABBITT METAL — TIN- & LEAD-BASE. When non-ferrous mining technique produced metals such as tin, lead, antimony, and zinc in volume and at low price, the modern trade-name soft bearing metals which were originally pioneered by Isaac Babbitt came into general use.

These materials are broadly referred to as tin-base or lead-base since all modern soft bearing metals consist primarily of either of these two base

Table I Copper-base bronzes. (Taken from "Bearing Metal Data," courtesy of Federal-Mogul Corp.)

Metal	Average Composition	Relation to Lubrication
F.1 - "Admiralty Bronze" or "Gun Metal" SAE Spec. 62	Copper 88% Tin 10% Zinc 2%	To be used where lubrication is very good, or motion is of a rocking nature. A good gear bronze suitable for heavily loaded piston pin bushings, or similar purposes.
F.2 - 80-10-10 leaded bronze. SAE Spec. 64	Copper 80% Tin 10% Lead 10%	One of the strongest and hardest of the lead bronzes. Should not be used where lubrication is poor. Where properly lubricated it is adaptable to heavy loads and severe usage.
F.3 - "Ounce Metal", "Steam Metal" or "Red Brass." ASTM Spec. B, 62-28. SAE Spec. 40	Copper 83-86% Tin 4-6% Lead 4-6% Zinc 4-6%	In bearing field used largely as backing for babbitt lined bearings.
F.5 - Known as SAE No. 66 Alloy	Copper 85% Tin 5% Lead 9% Zinc 1%	Used for babbitt lined bearings backs and for bushings where service is not severe. Due to low resistance to pounding should not be used under heavy loading.
F.6 - "Ex-B Metal", "Engine Brass" or "Semi-Plastic Bronze" SAE Spec. 67	Copper 76-78% Tin 7-9% Lead 14-16% Zinc 0.5% Max.	Suited to average bushing applications but requires better lubrication than metals with a higher lead content.
F.8 - SAE Spec. 660	Copper 83% Tin 7% Lead 7% Zinc 3%	A good average bearing bronze, suited to medium loads. Lacks resistance to pounding due to low strength and hardness.
F.11 - "Leaded Gun Metal" or modified "Admiralty Bronze" SAE Spec. 63	Copper 87-89% Tin 9-11% Lead 1.5-2.5% Zinc 1% Max.	A general utility bronze adaptable to heavy loads and severe working conditions. Must be well lubricated, however, or overheating and scoring may develop. By reason of its lead content the wearing and anti-friction properties are better than straight admiralty bronze.
F.12 - A modification of "Journal Bearing Metal" F.16	Copper 71-75% Tin 6-8% Lead 18-20%	Suitable for moderate to heavy load bushings. Has good resistance to pounding out, cracking or scoring. One of the best high lead alloys.
F.15	Copper 70% Tin 10% Lead 20%	Suitable under adverse lubrication conditions, and moderately heavy loads due to high lead content.
F.16 - "Journal Bearing Metal"	Copper 70% Tin 5% Lead 25%	Useful where lubrication may be interrupted. Under low speeds and light loads may even function for some time without lubrication. Soft steel shafts will not be scored should lubrication fail except under very high speeds or low clearances. Very low in resistance to shock or pounding.
F.19 - "Ford Z Metal"	Copper 85-89% Tin 7.5-8.5% Lead .75-1.5% Zinc 2.25-4.25%	A strong ductile alloy of average hardness suited to service to which other low lead compositions would be adapted.

metals, with possibly lesser quantities of zinc and some other element or elements such as antimony, arsenic, or copper to impart additional strength and hardness for special service applications. The more common types are summarized in Table II.

Tin-Base Metals. Bearing metals of this type contain from 65 to 93 percent of tin, 3 to 10 percent of copper, and 3 to 15 percent of antimony. Up to 5 percent of lead also may be added to improve the anti-friction properties. Nickel also is sometimes added in small quantities to improve the hardness. The tin-base metals are in general the hardest of the so-called Babbitt materials. Their toughness and tensile strength are features which render them well suited for service involving constant pounding action as would prevail in the connecting rod bearings of high speed engines. They have good resistance to corrosion but their fatigue resistance is lower than in the lead-base metals.

The addition of zinc to a tin-base metal produces a bearing which has very high compressive strength. This is a desirable feature in bearings which are subjected to an extremely heavy pound, as in bolt heading machines and certain types of rock crushers. Lubrication must be positive and dependable, otherwise a metal of this type may seize and cut the shaft.

Lead-Base Metals. Lead-base metals contain from 70 to 91 percent of lead, 8 to 20 percent of antimony, and 0.5 to 18 percent of tin. Some copper also may be added as replacement for some of the tin content, although the inclusion of this metal requires special handling in the original alloying procedure. When this is done it is practicable to replace as much as 27 percent of the tin content with copper.

Lead-base metals have good embeddability, fair hardness and compressive strength, good fatigue resistance, low tensile strength, and elongation. They have the advantage of excellent self-lubricating qualities and low first cost. On the other hand their relatively low resistance to corrosion must not be overlooked.

COPPER-LEAD BEARINGS. It has been noted earlier that copper-lead as a bearing material is a product of fairly recent development. In reality its technical development and wide acceptance can be credited to the expansion of the automotive and internal combustion engine industries. When engines were designed to develop increased power, the ability of the bearings to carry the load became an important factor. Flood lubrication of engine bearings by oil was used from the beginning but full fluid film lubrication could not carry the load all by itself. The answer was the copper-lead bearing due to its load-carrying capacity. The more common types are summarized in Table III.

Copper-lead bearing materials in general will contain 20 to 35 percent of lead, the balance being chiefly copper. Some tin can be added to increase the strength and reduce the cracking tendency. A small percent of silver also is said to improve the microstructure and prevent lead segregation.

When copper-lead bearings first were introduced, corrosion presented quite a problem due to attack by the oils available at the time. Usually the lead was preferentially removed, but, at low temperatures and in the presence of water the reverse was true and the copper was removed. Intensive research, however, proved that corrosion-resistance could be markedly improved by use of special inhibiting additives. This paved the way for

Table II Typical tin-base and lead-base bearing alloys (Babbitt metals). (Courtesy of A. W. Cadman Mfg. Co.)

Metal	Composition Range		Relation to Lubrication
High Tin-Base Babbitt. Grades 1 to 4 incl. Federal Spec. QQM161a	Tin Antimony Copper	80.5 to 93.0% 3.5 to 14.0% 3.5 to 8.5%	Such alloys, particularly those containing 88 to 90 percent tin, and antimony approximately twice that of copper, are suitable for high speeds and heavy loads including shock or reciprocating loads; of course under favorable lubrication conditions. Oil should be used of a viscosity in accordance with the clearance between the bearing lining and the shft.
High Tin-Base Babbitt. Grades 1 to 3 incl. ASTM Spec. B23-49	Tin Antimony Copper	83.0 to 92.0% 4.0 to 13.0% 4.0 to 8.5%	Same remarks as above.
Tin Babbitts. Grades 1 to 4 incl. Navy Spec. 46M2F	Tin Antimony Copper	80.5 to 93.0% 3.5 to 14.0% 3.5 to 8.5%	Same remarks as above.
High Lead-Base Babbitt. Grades 10 & 13. Federal Spec. QQM161a	Lead Antimony Tin Copper Arsenic	78.0 to 88.0% 8.0 to 17.5% .75 to 17.5% .4 to .6% .2 to 1.4%	Modern alloys within this range, particularly those containing lead, antimony, arsenic and copper at or near the high limits, and tin of 1.4 percent maximum, are designed for operation at high speed, heavy load, and slow speeds along with extremely heavy loads. Either oil or grease can be used, but where slow speeds and light loads prevail and insufficient frictional heat is developed to adequately soften the grease, the use of calcium soap-base greases should be avoided.
High Lead-Base Babbitt. Grades 10, 11 & 15. ASTM Spec. B23-49	Lead Antimony Tin Copper Arsenic	82.0 to 91.0% 14.0 to 17.5% .75 to 2.25% .50 to .60% .20 to 1.40%	Same remarks as above.

the modern heavy-duty motor oils which are so effective in protecting the bearings of present-day automotive and internal combustion engines.

POROUS BEARING METALS. The science of powdered metallurgy led to the development of the porous bearing metals. The manufacturing process involves moulding of the desired alloys proportioned according to the intended service, heat treating or sintering to set the component metals, and impregnating with a light to medium viscosity oil of SAE 10 to 20 grade. Conventional engine or motor oils of good resistance to oxidation and gumming generally are preferred within the 40 to 80 centistoke at 100 F. viscosity range.

Non-ferrous metals of this type are of the nature of heavy duty bronzes. Powdered aluminum and iron also are used for certain special purposes. A feature of the powdered bearing metals is their ability to absorb about one third of their weight of oil. In service this oil gradually exudes to the bearing surface to furnish dependable lubrication for a considerable period of time. The heavy-duty porous bronzes and ferrous alloys are claimed to have a load-carrying capacity comparable with that of similar solid metals.

Installation of such metals in a journal must be carefully done. Machining to fit is not recommended because the porosity of the surface may be harmed by the cutting tool. For this reason it is best to specify the exact size of the bearing, taking into consideration the desired clearance when ordering.

Due to the comparatively low ductility of such metals their use is restricted both as to speed and service. When adequately lubricated they can carry loads comparable to solid metallic bearings, but excessively low or high speeds can break the oil film. Continued operating temperatures much above 150 F. can have the same effect.

Table III Copper-lead bearing alloys. (Courtesy of Cleveland Graphite Bronze Co., Div. of Clevite Corp.)

Metal	Nominal Composition	Characteristics
F-12 SAE 480	Copper 65% Lead 35%	Cast on low-carbon steel-back. Medium to heavy duty bearing applications. High lead content provides adequate surface action. Hardened shaft desirable.
F-14 SAE 48	Copper 71% Lead 28% Silver 1%	Cast on steel shells. Medium to heavy duty bearing applications. Frequently used in turbo-superchargers, aircraft reduction gears and other high speed or high load applications with plated overlay. Hardened shafts desired. Has excellent combination of high fatigue resistance and surface action.
F-42 Similar to SAE 481	Copper 60% Lead 40% Tin .3%	Cast on low carbon steel. Medium duty. Used as resizable replacements for thin-babbitt bearings. Soft shafts satisfactory.
F-77 SAE 49	Copper 75% Lead 24% Tin .25%	Cast on low carbon steel back. Heavy duty bearing applications. Requires overlay, positive lubrication and hardened shaft. Excellent fatigue resistance. Widely used in automotive and tractor applications.

RUBBER BEARINGS. Rubber as a bearing material is adaptable to any under-water location or where water can be piped to the bearing for lubrication purposes. The hardness of the rubber used is comparable to the hardness of an automobile tire. With a well-lubricated rubber bearing, the coefficient of friction is about the same as for a well-lubricated metal bearing; it is especially low at higher speeds. The most usual applications of rubber as a bearing material include marine propeller installations, rudder pintle heads, dredge cutter heads, deep well pumps, coal washers, under-water conveyors, and hydraulic turbines. In such service there is relatively little possibility of the bearing material coming in contact with petroleum oils; as a result natural rubber after suitable vulcanizing or any of the synthetic rubbers can be used. Where there might be possibility of oil contact, however, thickol or one of the nitrile rubbers which are most resistant to volume change should be used. Nitrile rubbers are more easily wet by water and thus maintain an effective lubricating film at higher speeds and loads.

Provided the type of rubber in the bearing is suited to the service conditions, it affords a definite advantage as a protective medium for metal shafting by reason of its ability to accommodate itself to abrasive particles. In other words, the softness of the rubber enables any foreign abrasive solid matter, carried in by the water, to be depressed into the bearing surface, rolled into the longitudinal grooves, and flushed out rather than being imbedded in the surface of the bearing, as might occur with some metal bearings, to cut or score the shaft. The shaft should be bronze, "monel" metal, or stainless steel because steels of conventional type present a rusting problem especially when the machine is idle. Rubber is possessed of another natural advantage by reason of its flexibility which enables it to conform to some shaft misalignment without undue vibration.

While it has been noted that the coefficient of friction of a well-lubricated rubber bearing is low, this can be high unless the bearing is copiously lubricated by water from the very start. Even a few turns of the shaft on a dry bearing surface could cause serious abrasion, tearing, and over-heating. It is well, also, to bring the unit up to speed before the load is applied because even though rubber, as a bearing material, can accommodate itself to loads around 50 lbs/sq.in., temperature control and complete lubrication are necessary to protect the surface. The normal maximum shaft speed limit for such bearings is around 800 to 1000 ft./min. The normal accepted maximum permissible temperature is approximately 150 F. Higher temperatures will cause marked decrease in life. Shaft sizes in hydraulic turbines may range up to 37½ inches in diameter.

Most complete protection of the bearing by water lubrication is assured when the bearing is designed to insure complete flooding by the flow of water. Chamfering of bearing edges and angular

grooving enable the turning shaft to exert a pumping action, just as it does in a well designed oil-lubricated metal bearing. Flood lubrication also permits the water to function not only as a lubricant but also as a means for conducting heat and foreign materials away from the bearing.

ALUMINUM ALLOYS. Aluminum alloyed with approximately 7 percent of tin, plus some silicon, silver, nickel, cadmium, or copper, gives a bearing material of excellent heat conductivity and resistance to corrosion, plus high load-carrying capacity. Steel backing is necessary to obtain the benefit of the high inherent yield and fatigue strength of aluminum, but bonding to steel must be carefully done to assure permanence; otherwise there may be a tendency for the bearing to loosen in service.

The relative lightness of aluminum bearing alloys seems to promote their ability to retain an oil film. This feature and the high heat conductivity is beneficial in case of hot spot occurrence and the necessity for adequate cooling to prevent bearing damage. There is no restriction as to type of lubricant with an aluminum alloy bearing. Conventional motor oils, engine oils, or greases, all can be used within the viscosity or consistency ranges normally employed with bearings of a given size or speed.

SILVER & SILVER ALLOYS. Silver overplated on steel, or bearing metal of usually copper-lead base, is another type which warrants mention, although the relative cost is high as compared with other more conventional bearing materials. Overplating is done by the electroplating process. Silver as a bearing material has been used chiefly in aircraft engine service; on the other hand, silver alloyed with cadmium as a base, as cadmium-silver, is applicable to industrial machinery. The high cost factor of both metals, however, is a deterrent to extended use.

High load-carrying ability and resistance to fatigue are advantages possessed by silver, but to offset this is the low so-called "wetting ability" or ability of oil to adhere to the surface.

GRAPHITED BEARING METALS. Bearing metals of this type are bronzes or white metals incorporating graphite. The latter serves as a lubricant, so re-lubrication is required far less frequently than with straight metallic materials. In fact, such a bearing metal is virtually self-lubricating. Due to the fact that most of these mixtures are porous, their load-carrying capacity is limited to around 600 psi. On the other hand, the graphited copper-base material is an exception and can be used for high speed up to around 2500 ft./min. and high temperature to around 700 F. Babbitt, bronze, and cadmium base graphited alloys involve somewhat lower maximum operating temperature ranges. Load-carrying ability varies inversely with speed, so with low loads the top speed is safely attained.

NON-METALLIC PHENOLIC-BASE MATERIALS. With bearings made of a fabric base which is laminated and bonded with phenolic or resinoid thermosetting or thermoplastic material, water is usually the most satisfactory lubricant. However, oil can be used satisfactorily if desired; in fact when the machine is to be idle for any length of time it is well to provide for delivery of a petroleum base light oil or grease just before shut-down to prevent rusting of the shafting.

Phenolic materials are highly resistant to moisture, have high mechanical strength and ability to carry heavy loads, are not affected by corrosive materials or refinery products, resist stray currents due to the high dielectric strength, have low coefficient of friction, and do not deteriorate with age.

The water generally used as the lubricant must be free from abrasive materials, as otherwise the bearing surface may become scored and the desired low frictional benefits will be reduced. The quantity of water required will vary according to the speed, pressure, and inlet water temperature. Five to ten gallons per hour for each square inch of projected bearing area will usually give ample protection. A good flow of water is necessary, especially where such bearings are heavily loaded, since materials of this nature do not conduct heat readily and bearing temperature control is almost entirely a function of the lubricating system.

NYLON. Nylon moulding powders proved suitable as bearing materials when their high tensile and compressive strengths and their resistance to heat and abrasion were realized. Nylon is comparable to the non-ferrous bearing materials in that it can be moulded to the shape of the shaft. This is done by injection into moulds of the desired size and shape. In the finished form the surface can be ma-

Table IV Operating characteristics of sleeve bearing materials. (Note: These figures must of course, be regarded as approximate due to the relation of type of material to method of lubrication and the prevailing operating conditions. In fact, some authorities contend that with satisfactory lubrication, the speeds indicated above, for example, often can be far exceeded.)

Material	Average Load Carrying Ability lb./sq./inch	Usual Maximum Operating Temp. °F.	Usual Maximum Operating Speed, ft./min.
Wood	up to 2000	150 to 200	2000
Bronze	up to 3500	350	1000
Babbitt			
Tin-base	up to 1800	250	4000
Lead-base	up to 1500	270	4000
Copper-lead	up to 3500	300	4000
Porous Metals	up to 4000	150	1500
Rubber	up to 50	150	1000
Aluminum Alloy	up to 4000	225 to 250	1500
Silver	up to 4000	300	4000
Graphitic Materials	around 600	up to 700	2500
Non-Metallic Phenolic-base	up to 4000	200	2500
Nylon	up to 1600	250	500

chined if desired. An added feature is that since the material is capable of embedding any abrasive particles which may be carried in by windage or by the lubricant, the bearing surface normally is not affected and danger of shaft scoring is reduced.

The heat-resisting characteristics of this bearing material also are of interest. Nylon is safely usable at continued exposure to a temperature range of 250 to 300 F., provided: (1) adequate lubrication is available, and (2) there is adequate space to allow for expansion which would reduce the clearance. An SAE No. 10 to 30 motor oil usually is the preferred lubricant according to the operating loads and temperature range. On the other hand, water, milk, petroleum solvents, or a variety of other liquids can be used where these liquids are in contact with the bearings as is the case in some types of pumping operations. The resistance of Nylon to solubility in alkalis or most dilute mineral or organic acids is a factor in this regard.

SELECTING THE MATERIAL. To assure that the best bearing material is selected and that it is fitted to the job, the most careful attention should be given to the operating conditions under which it later must function. Load, speed, and temperature must be considered particularly as to the extent to which any or all may vary. Bearing manufacturers take these factors into consideration when studying the component characteristics of their products. The science of bearing material production coordinates the finished product with the operating conditions. It is not enough for the purchaser to specify only the components in a bearing or its chemical make-up. How they are combined to make a finished bearing is equally as important to the ultimate successful performance as their basic characteristics. The tabular data contained in this brief review are indicative of the large amount of research which has accompanied the problem of tailoring the bearing to the job.

Section News

N. CALIFORNIA, September. John Boyd (ASLE President), Westinghouse Electric Corp., presented a paper entitled "Sliding Type Bearings," illustrated by a color film showing model bearings in operation and followed by a discussion on "Pivoted Pad Bearings." Pictured above are Mr. Boyd (left), and R. H. Decker (Program Chairman), Tide Water Associated Oil Co.

October. H. L. Hemmingway, Sales Service Depts., Pure Oil Co., presented a paper entitled "More Power To You." (Submitted by G. H. Hommer, Reporter, and A. C. West, Sec'y-Treas.)

BALTIMORE, September. Dr. W. E. Campbell, The Brush Laboratories, Div. of Clevite Corp., presented a paper entitled "Fretting Corrosion."

October. L. E. Hoyer, American Brake Shoe Co., Railroad Bearing Laboratory, presented a paper entitled "The Laboratory and Its Part in the Hot Box Problem." (Submitted by J. E. Buchanan, Sec'y.)

BOSTON, October. Prof. M. C. Shaw, Massachusetts Institute of Technology, Machine Tool & Metal Cutting Div., presented a paper entitled "Grinding Fluids for Titanium," illustrated with slides and followed by a Question & Answer period. (Submitted by E. B. Harvey, Sec'y.)

BUFFALO, September. M. Ehrlich, American Lubricants, Inc., presented a paper on the subject "Hi-Temperature Greases."

October. F. Kenline, Lake Erie Engineering Corp., presented a paper on the subject "Applied Hydraulics." (Submitted by L. E. Locke, Jr., Sec'y.)

CHICAGO, October. A "Stump the Experts" session with W. P. Youngclaus, Jr., ASLE Administrative Secretary, acting as moderator. Panelists included: P. V. Toffoli, Pure Oil Co.; C. B. Robb, U. S. Steel Corp.; H. H. Curtis, D. A. Stuart Oil Co.; and W. C. Bryant, Magie Bros. Judges included: J. W. Peterson, Standard Oil Co.; C. C. Blaidell, Penola Oil Co.; and E. J. Newton, Van-Straaten Chemical Co. (Submitted by A. B. Wilder, Sec'y-Treas.)

FORT WAYNE, September. R. McMahan, Socony-Vacuum Oil Co., an authority on developing fuels and motor oils, and a technical consultant for the Indianapolis 500-mile Speedway races, showed a color film of the 500-mile race together with slides of the race cars in the various stages of assembly. Following this presentation, Mr. McMahan discussed the "behind-the-scenes" happenings covering the changes and improvements which have made possible the safety and operational features of the present day car.

October. T. J. Christiansen, Dow Chemical Co., presented a paper on the subject "Solvents, Their Application & Handling in Industry," followed by a kinescope presentation of the first television broadcast of the show "Medic"—an unforgettable showing of the birth of a baby. (Submitted by J. W. Buckner, Program Chrmn.)

KANSAS CITY, October. A 3-day Course in Lubrication was held at the Aladdin Hotel encompassing the following sessions: "The Principles of Lubrication,"

(Continued on p. 351)

TAXICAB ENGINE WEAR & CLEANLINESS

by O. W. Wuerz*

Fleets of 1600 to 2000 taxicabs were studied to determine causes of top-cylinder wear and sludge formation. Based on conclusions derived from these studies, a high additive-content detergent oil was indicated. Application of this oil to fleet operation proved highly successful. Selection of an oil to satisfy a given operation is discussed. It is concluded that an oil Supplement 1 level additive-content is justified in taxicab service.

Taxicab operation is, in general, "stop and go" driving with frequent acceleration and a big percentage of idling. Under the API classification it is M.S. service. A taxicab will travel between 50,000 and 75,000 miles per year. It will be "working" as much as 20 hours per day with the engine idle from 30% to 50% of the time the cab is on the street. Average speed is low, in the order of 15 mi/hr. Traffic conditions cause an almost continuous cycle of acceleration and braking.

Maintenance schedules are based on lubrication requirements, periodic tightening and adjustment, unit replacement and tune-up. In general, maintenance schedules of individual fleets will vary depending upon the preference and ability of the supervisor. However, once a schedule has been established it may not lightly be changed.

Engine over-haul is a major item with the fleet operator as it involves the threefold cost of lost revenue, mechanics' time, and either new parts or the cost of reconditioning used parts. The need for over-haul is seldom dictated by power requirements, but it is most frequently determined by oil consumption or excessive blow-by which may seep into the cab and cause driver or passenger complaints. The immediate cause of these conditions are high top-cylinder wear and excessive sludge formation.

Data were taken from fleets of 1600 to 2000 taxicabs operating in New York City and cover a period of more than 20 years. They are mainly statistical in that they are fleet averages or are the result of large numbers of measurements or observations made on engines in regular taxicab service. This paper covers experience in operating large fleets of taxicabs on straight mineral oils and several additive oils. It discusses the desirability and advantages of supplying large fleets of taxicabs with high additive oils.

DESCRIPTION OF FLEETS. Pertinent data regarding the various fleets concerned in this study of engine lubrication are given below:

Fleet Year	No. of Cabs	Ring Description	Thermostat Temperature
1933	2000	5 rings, cast-iron	140 F.
1936	2000	"	160 F.
1940	2000*	"	180 F.
1948	1600	"	160 F.
1950	1600	"	160 F.
1953	1600	5 rings, chrome top ring	160 F.

*This fleet reduced to 1600 during war years.

All fleets were powered by In-line, L Head engines and all used 5 ring pistons. Radiators were equipped with automatic winterfronts controlled by thermostats as indicated above. Hardness of the block is not a factor in the results obtained because all engine blocks since 1940 met the same hardness specification.

OILS USED. With the exception of the 1933 fleet, all fleets were supplied with substantially the same type of mineral oil up to 1950 when a high additive Supplement 1 oil was used exclusively in that fleet.

Fleet Year	Oil	Viscosity Grade	Specification
1933	Straight Mineral	Winter SAE 30 Summer SAE 40	Mid-Continent
1936	Straight Mineral	Winter SAE 30 Summer SAE 30	500 Pale Oil
1940	Straight Mineral	Winter SAE 20 Summer SAE 20	300 Pale Oil
1948	Straight Mineral	Winter SAE 20 Summer SAE 20	300 Pale Oil
1950	Straight Mineral	Winter SAE 20 Summer SAE 20	Supplement 1 or equivalent
1953	Additive	Winter SAE 20 Summer SAE 20	Supplement 1 or equivalent

Note that there was no change in viscosity specifications from Summer to Winter starting in 1936. Oil specifications were written to assure a type of oil and not a particular characteristic.

STRAIGHT MINERAL OIL EXPERIENCE. Originally only straight mineral oils were available and, in general, they were adequate. However, a study of the oil consumption record of the 1933 fleet developed several interesting features: One thousand cabs were put into service during the early part of that year and the second 1000 cabs during the latter part of the year. The first group had carburetors with vacuum controlled accelerating systems and the second group had mechanically actuated accelerating systems. The fuel consumption of this second 1000 cabs was almost exactly 1.0 miles/gallon greater than the first 1000.

The significant features of these data are that,

*Cab Service & Parts Corp., 315 W. 68th St., New York 23, N. Y.

This paper was presented at the ASLE 9th Annual Meeting, Cincinnati, April 7, 1954.

immediately following September of any given year, oil mileage decreases sharply. There is a further drop in oil mileage during the winter months and then it remains almost constant throughout the summer. Changes in oil consumption are more pronounced for the second 1000 cabs but follow the same general seasonal pattern as the first 1000. Apparently atmospheric temperatures and also the rate of fuel consumption produce engine changes which affect oil consumption. A more detailed investigation of the next fleet was planned.

Top-cylinder wear appeared to be of greatest significance and it was therefore determined on a large number of engines of the 1936 fleet. Measurements were made across and lengthwise of the engine for each cylinder at the following locations: above ring travel, after removing deposits; at top of ring travel, one-half inch below and one inch below. The average of all cylinders was reported as the wear for that engine at the designated location. Cylinder wear was determined on many engines of this fleet, selected at random. Fig. 1 shows cylinder wear of the 1936 fleet at the top of ring travel and one-half inch down. For the first few thousand miles, the rate of wear is high and is attributed to initial break-in. Thereafter the rate became stabilized at a lower value which held to about 40,000 miles.

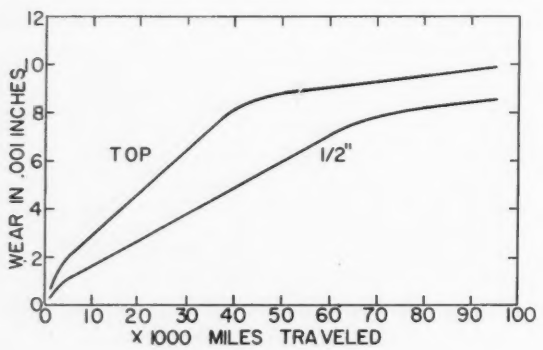

Fig. 1 Showing the progress of cylinder wear at the top of ring travel and ½ inch below in taxicab engines when supplied with a straight mineral oil.

The oil consumption record for the 1936 fleet was substantially the same as that already discussed for the 1933 fleet.

Returning to the 1936 fleet, the average miles traveled per month per cab was about 3,500. After the cabs had been in service an average of 11 months, the oil mileage dropped from about 2,000 to about 500 miles/gallon.

Even though the rate of oil consumption increased constantly over these 11 months, or during 40,000 service-miles, the rate of top-cylinder wear remained constant. It is concluded that top-cylinder wear in taxicab service is not a function of the quantity of straight mineral oil that is on the engine surfaces swept by the rings.

After 40,000 miles, at which time the average wear is about .0085", there is a short transition period after which the wear rate again becomes constant but at a much lower value.

Ring wear was not measured as a regular procedure. Spot checking at the time of re-ringing showed a top compression-ring gap increase to .070" or more. It is concluded that, with the top-cylinder and the ring worn as indicated, the top compression ring did not exert sufficient pressure against the wall either to control oil or to remove material from the cylinders.

The economics of taxicab operation did not justify re-boring cylinders. Therefore at engine over-haul new rings and, as required, new pistons were installed. For a short period after re-ringing, the rate of wear at the top of ring travel approached the rate obtained at less than 40,000 miles. At about 15,000 miles after re-ringing, oil consumption again increased and the wear rate decreased in the same manner as shown at the 50,000 mile point for the original rings in Fig. 1. Progress of wear at the half-inch level is very similar to that at top of ring travel except that the initial rate is lower and the transition period is longer.

Wear at the bottom of the ring travel was small, about .002" to .003" at 90,000 to 100,000 miles, and was greatest over the thrust area of the cylinder.

Visual inspection of these engines showed that maximum top-cylinder wear occurred over approximately one-quarter (1/4) to one-third (1/3) the circumference of the cylinder. This area of maximum wear was not necessarily that which took the piston thrust. Two lines drawn from the intake valve, one to the front and the other to the rear of the cylinder, would generally designate the area over a portion of which maximum wear would occur. Rarely did high top-cylinder wear develop on the valve side.

In this fleet little or no trouble was experienced with sticking rings. Deposit build-up behind the rings did not reach an amount where the rings could not be pressed into the groove below the surface of the ring lands. Evidently the mechanism producing wear at the top of ring travel is different from that which produces wear at the bottom of the cylinder.

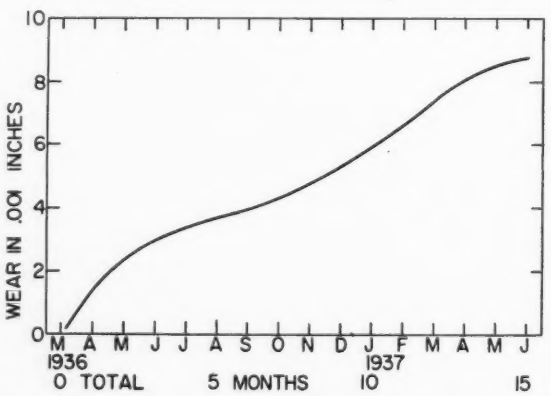

Fig. 2 Illustrating the seasonal effect on cylinder wear at the top of ring travel on taxicab engines using straight mineral oil.

SEASONAL EFFECTS ON WEAR. All cabs in the group to be discussed were put into service during March and April of 1936. Cylinder wear of this group of engines was arranged as a function of months of service instead of the more usual miles traveled. These data are presented in Fig. 2. There is evidently a relationship between cylinder wear and atmospheric temperature. The middle portion of this curve is of major interest: The first few months of operation represent the break-in period during which wear was abnormal; the last portion represents the period where top rings were no longer functioning properly. It is therefore disregarded in this analysis.

The data from July 1936 through March 1937 may be described by the equation

$$y = .0021e^{.104x} + .0004e^{-.190x} \qquad (1)$$

where y = average cylinder wear at top of ring travel, x = months of service. This equation is valid for x greater than 3 and less than 13.

The effect of atmospheric temperature on the rate of wear is to be determined. The first derivative of equation 1 gives the slope or in other words, the rate of wear

$$dy/dx = .000218e^{.104x} - .000076e^{-.190x} \qquad (2)$$

Calculating the rate of wear per month of service, these values are plotted against the mean monthly temperature for the corresponding month. These results are shown in Fig. 3. Note that the rate of wear increases moderately as the temperature decreases to about 45 F., at which point the curve breaks and the rate increases sharply. Cylinder wear is apparently determined by some process which is controlled by, or related to, atmospheric temperature.

During 1940 another fleet was put into service. The automatic winterfront was controlled by an

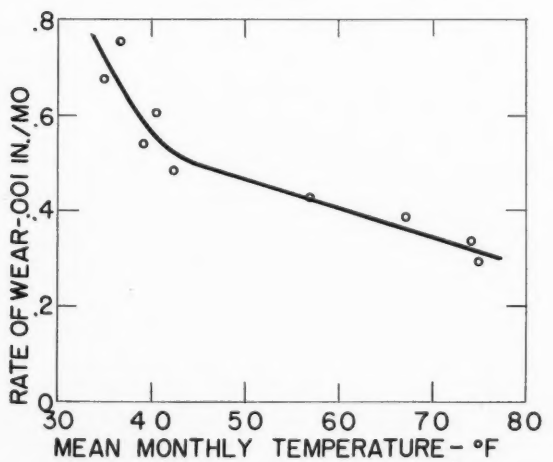

Fig. 3 Showing the effect of ambient temperature upon the rate of top-cylinder wear on engines in taxicab service when supplied with straight mineral oil.

180 F. thermostat and the engine was lubricated with a straight mineral oil of SAE 20 grade. Because of the war this fleet was operated far beyond its anticipated life and, for the same reason, we measured approximately only 100 engines. The wear curve as shown in Fig. 4 exhibits the same general characteristics that the 1936 fleet did.

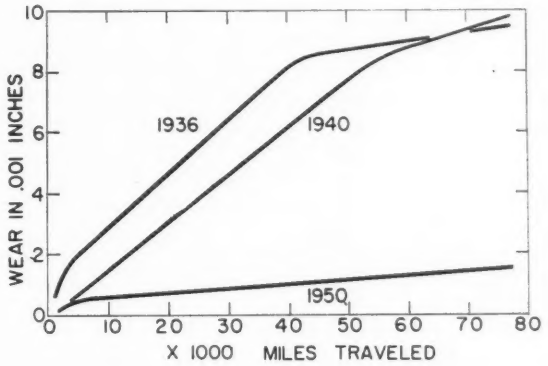

Fig. 4 Illustrates the magnitude of top-cylinder wear of engines in taxicab service when supplied with Supplement 1 additive oil and using hard chrome-faced top rings. Comparative curves for engines using straight mineral oil and cast-iron rings are shown.

Specifically, ring life was extended about 15,000 miles and the rate of wear in inches per 1000 miles was .00016 as against .00018 in 1936.

The last fleet to be supplied with a straight mineral oil was put into service during 1948. Because of the extensive data on cylinder wear in taxicab service already available, no attempt was made to make the large number of wear measurements taken on previous fleets. Enough engines were checked to determine the trend and magnitude of cylinder wear. This was found to agree in all respects with previous fleets.

Heavy sludge formation is a characteristic of taxicab engine operation. In the era of straight mineral oils many methods of alleviating this condition were proposed. Among these were: use of oxidation resistant oils, increasing engine temperatures, insulating oil pans, use of oil filters, etc. No proposal was economically feasible. The best merely extended the time interval over which sludge accumulated. The cost of installation or the cost of maintaining it could not be justified. Early in the study of engine sludge it was concluded that it was mainly a product of the fuel and not of oil deterioration. The major harmful effects traceable to sludge were: clogging of oil rings, sticking of rings in grooves, and eventual clogging of main oil-galleries. Since sludge could not be prevented from forming and depositing in the engine, it was decided to develop an inexpensive method of periodic sludge removal. Starting with the 1940 fleet, the use of 50% tar-acid oil for de-sludging engines was adopted. This material, consisting principally of crude cresilic acids in coal-tar neutral oil, was a highly effective solvent for engine sludge. Later

the tar-acid oil was fortified with furfural. This combination gave a most effective solvent action. The procedure for de-sludging, or flushing engines, was tailored to fit into the preventive maintenance program. This was accomplished at no increase in the over-all service-time. This procedure was used until the 1950 fleet was put into operation. At that time the high additive-content oils provided a more satisfactory method of maintaining commercially clean engines.

This discussion thus far has developed the following:

Top-cylinder wear was the factor limiting engine life in taxicab service.

The amount of straight mineral oil present on the cylinder surfaces did not affect the rate of wear at the top of ring travel.

Atmospheric temperature markedly influences cylinder wear.

Maximum wear does not necessarily occur over the thrust area of the cylinder.

The rate of fuel consumption affects oil consumption and, by deduction, fuel consumption also affects cylinder wear. (Recall the oil record of the 1933 fleet.)

The operating temperature of the thermostat affects cylinder wear. The thermostat controls cooling-system temperatures and also carburetor air-intake temperatures.

The mechanics by which wear at the top of ring travel is created are different from those producing wear at the bottom of the cylinder.

SELECTION OF ADDITIVE OIL. The amount of additive in an oil necessary to meet a given requirement is not always readily determined without a thorough study of operating conditions.

The following factors must be brought into satisfactory balance: (1) additive content of oil, (2) type of service, (3) drain intervals, (4) rate of oil consumption, (5) engine characteristics, (6) fuel, (7) cost.

There are, in general, four levels of additive-content oil from which to choose. There is the old 2-104B, a now obsolete specification but oils are available at about this additive level. These oils could be given consideration for M.L. for M.M. service. The Mil-0-2104 has a considerably higher additive content and would be considered for M.M. or M.S. service. 2-104B Supplement 1 specifiication generally refers to oils having higher additive content but some Mil-0-2104 oils meet this requirement. Although this specification is also obsolete, equivalent oils are available. Finally, Series 2 oils with extremely high additive levels are available. These are suited for DS and possibly M.S. service. A consideration of the type of service can therefore narrow the choice to one principal type of oil.

The next factor is required oil-service miles or the length of the drain period. Naturally the longer the drain period the greater the amount of additive required.

The next item to consider is the rate of oil consumption of the engine. During the combustion process, acids and water vapor are produced. The acids will vary in nature and amount depending upon the composition of the fuel, the fuel-air ratio, and distribution to each cylinder. To completely protect the engine against corrosive cylinder wear, there must be available sufficient alkaline material to neutralize all the acid products of combustion. Thus, for an oil with a given additive content, consumption must be at a rate which will form a film of oil on the cylinder walls containing the required amount of alkaline material to neutralize these acids completely. A higher rate of oil consumption than the optimum is wasteful and a lower rate will give something less than complete protection.

The fifth factor is engine characteristics. Some of the factors which will influence the desirable additive content of the oil are: manifolding and distribution, engine operating temperatures, circulation of coolant, and fuel-air ratios. The over-all effect of these factors may be roughly approximated by the quantity and type of sludge which is deposited on engine surfaces. A heavy deposit of mayonnaise-type sludge in valve chambers, oil pans and crankcases, and on oil screens indicates a high additive-content oil is required. Where additive oils are being tried, a careful study of these factors may indicate a minor engineering change which would make possible the successful application of those oils which would otherwise not be practicable.

Another item to be considered is the fuel. It is seldom that the fleet operator is in position to specify fuel properties and it is not suggested that suppliers be lightly changed. It is, however, suggested that the consumer study the effects of fuel upon the condition of his engines.

Fuel factors which can influence engine condition are volatility and selective distribution of the sulphur compounds. Ideally, volatility should be adjusted to the type of service. The oil companies inform us that this is impossible to accomplish. This is not a discussion of fuel characteristics but we are of the opinion that, when the consumer has made a careful enough study of his requirements and has available substantiating data, at least a workable compromise can be effected.

It is generally recognized that an increase in the 90% distillation temperature of the fuel will increase the crankcase dilution. At the same time crankcase ventilation is being improved. The vehicle cannot be moved with gasoline that dribbles past the rings and is either converted to sludge or exhausted through the breather. In the interests of the conservation of our national resources it is suggested that greater efforts be made to produce fuels of optimum volatility for maximum tank miles per gallon.

Distribution of fuel sulphur to the various cylinders may influence wear. A sulphur determination on a gasoline sample gave a value of .056%. A simple Engler distillation of this gasoline was made and three fractions were collected. Sulphur content was as follows: light distillate — .040%; middle distillate — .044%; heavy distillate — .110%.

A concentration of sulphur in one or more cylinders, brought about by poor distribution of low-volatility fuel, will raise the additive requirement of the entire engine to assure protection of that one cylinder. This factor must therefore be considered when deciding upon the additive level of the oil. In the event that observation or measurement of the engine shows a wide variation in wear between cylinders, a higher additive-content oil should be selected, at least for preliminary tests.

The final item to be considered is cost of the oil. It can be very simply stated that the cost per gallon is negligible compared to the benefits realized, providing the additive requirement of the fleet was correctly determined. In addition, drain periods must be rigidly maintained. If it is desired to change to an oil of different additive-content and cost, a complete re-evaluation of all factors is in order and adjustments must be made where necessary.

ADDITIVE OIL EXPERIENCE. The 1950 fleet was the first to be supplied with additive oil throughout its entire service life. A few hundred cabs received an initial filling of straight mineral oil as break-in oil. Fig. 4 shows the average wear at top of ring travel for the 1950 fleet. For comparison the same information is given for the 1936 fleet and for the 1940 fleet.

Top-cylinder wear was measured on three hundred and twenty engines of this fleet. The distribution of the data represented by the curve marked 1950 was as follows: 75.4% were within plus or minus .0005" of the curve, 5.0% were less than minus .0005" of the curve, 19.6% were greater than plus .0005" of the curve.

These cabs were under the direct control of the drivers for as much as 20 hours per day. In some instances a cab is assigned to a regular team for its service life; in extreme cases a different driver may take a cab every shift. Some drivers attempt to improve on the engineering, for example: blocking open winterfronts, or adjusting carburetors. Since these cabs were working for their living, it is possible that garage superintendents may, for various reasons, delay sending them to central shop for scheduled service. These are run-of-the-mill problems of large fleet operation and in most cases do not seriously affect the performance of the vehicle. A slight variation is expected.

Analyzing the 19.6% of the engines that showed high wear, and keeping the above considerations in mind, it was found that 13.1% will lie between plus .0005" and .0010" of the curve. The trend of wear for this group is normal; that is, it increases with miles and conforms to the general curve. No serious trouble would be anticipated. They would in all probability require re-ringing at lower mileage. Wear of the 6.5% group is abnormal. Speculation as to the causes would probably give the correct answer. However for the purposes of this discussion, this particular group may be disregarded. The 5.0% showing extremely low wear indicated good drivers and strict adherence to maintenance schedules. This group would represent the ideal; but it is not good business to attempt to hold a large fleet at this level of performance. Disregarding the very high wear group of engines, 80.8% of the data are within the plus or minus .0005" limits.

The results obtained with this 1950 fleet will be examined in more detail but at present the value, to the fleet operator, of engine condition as represented by the curves of Fig. 4 will be considered. At any time during the life of an engine, with top-cylinder wear of the order shown by the curve marked 1950, an effective ring-job can be done without reconditioning cylinders. Further economies are possible through re-using pistons which have not been cracked or broken. About 5% new pistons were all that we found necessary to install. This curve may be extrapolated to 200,000 miles which is greater than a taxicab will normally be in service. Engines have been checked at 150,000 miles or more as they were retired from service, and this linear relationship was found to hold.

Re-ringing is not determined by top-cylinder wear as was the case in previous fleets. In the 1950 fleet, ring wear, lower cylinder and piston wear were the criteria governing re-ringing.

The possible extension in original ring service-life is clearly shown in Fig. 5. The accumulated per cent of engines re-ringed is shown as the ordinate, and the abscissa as service miles at cell intervals of 5,000 miles. 96% of the 1948 fleet had been re-ringed between 65,000 and 70,000 miles, while only 34% of the 1950 fleet was re-ringed at equivalent mileage. After 18 months of operation 113% of the 1948 fleet had been re-ringed at less than 100,000 miles, but for a similar period only 62% of the 1950 cabs had been re-ringed at up to 120,000 miles. This record speaks for itself.

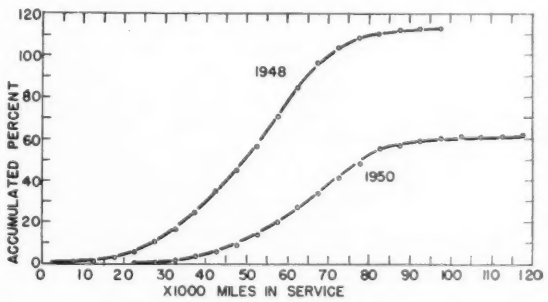

Fig. 5 Demonstrates the value in reduced maintenance as indicated by the number of engines requiring re-ringing when using high additive-content oil and chrome-faced top-rings as compared to the previous fleet using straight mineral oil and cast-iron rings.

As stated previously, besides the use of high additive-content oil the 1950 model cab had hard chrome faced top rings. In order to eliminate one variable, cast-iron top rings were installed in the engines in a group of 10 cabs as they were received from the factory. Top-cylinder wear for these engines is shown in Fig. 6. Note that again the same type of wear curve is obtained, i.e. a more

or less high rate of wear at first and then a sharp break to a low rate. The initial rate is, however, much less than was obtained with a straight oil.

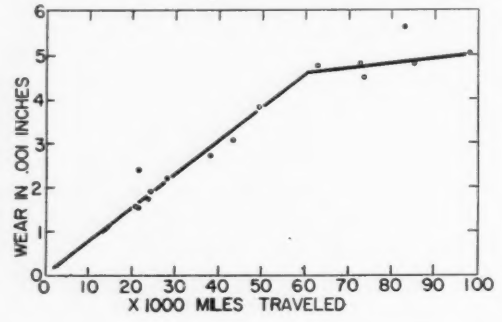

Fig. 6 Illustrates the magnitude of top-cylinder wear of taxicab engines when supplied with Supplement 1 oil and using cast-iron top-rings.

Table I gives quantitatively the effect of the various ring and oil combinations. Some reduction in cylinder wear can be obtained by maintaining a hot engine. Although the magnitude is not great, any improvement is acceptable to the operator. The benefit of chrome rings is appreciable, almost 34% reduction. The most outstanding benefit was obtained through the use of additive oil, in this instance 58.7%. The combination of additive oil with hard chrome rings reduced top-cylinder wear to a degree where this condition no longer determines engine over-haul.

Table I Variables affecting cylinder wear.

Table I	Rate Inches/1000 mi.	Percent Reduction in Rate
1. Straight Oil - C.I. rings	.000184	0.0
2. Straight Oil - C.I. rings*	.000156	15.2
3. Additive Oil - C.I. rings	.000076	58.7
4. Additive Oil - Chrome rings*	.000014	92.4
Effect of Chrome Rings		33.7
Effect of Additive Oil		58.7
*180 F. Thermostat		

In order to evaluate the effect of rate of oil consumption on cylinder wear, a group of 10 engines was re-ringed using all cast-iron rings which gave extremely severe oil control. Another group was re-ringed with cast-iron rings selected to give moderate oil control. Top-cylinder wear was determined on both sets of engines at various intervals, and the total make-up oil from ring installation to time of measuring wear was recorded. These data, as given in Fig. 7, show that the rate of top-cylinder wear is relatively constant at 0.00007 inches per 1000 miles up to the point at which the oil consumption equals one gallon per 3000 miles. Beyond this point the rate of top-cylinder wear increased with decreased oil consumption. Under the conditions of operation, oil consumption of 3000 miles/gallon supplied sufficient additive to neutralize acid products of combustion. At less than 3000 miles/gallon excess additive was being supplied. Above 3000 miles/gallon less than complete protection was being obtained.

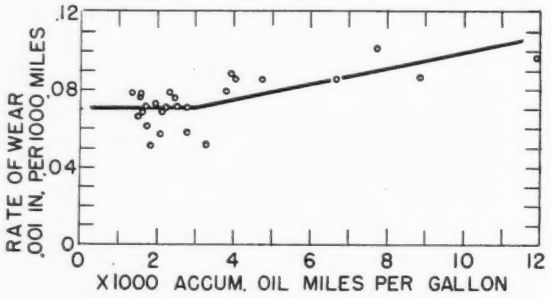

Fig. 7 Shows the effect of oil consumption on the rate of top-cylinder wear on taxicab engines supplied with a Supplement 1 level oil and using all cast-iron rings. The optimum oil mileage of the engines in question was found to be approximately 3000 M/G.

Commercially clean engines were maintained with the use of the high additive oil. Flushing with tar-acid oil and furfurol was discontinued. Used-oil analyses indicated that the amount of sludge formed was approximately the same as when straight mineral oils were used. However, in this case it was not deposited on engine surfaces but was held in suspension and drained with the oil. Valve chambers were satisfactorily clean. Sludge accumulation at this point is an indication that the oil is being continued in service too long and that the additive has been exhausted.

Table II Analyses of drain oil 1950 fleet. (Oil drained 2500-3000 miles.)

Table II Lic. No.	Oil Added qts.	pH	TBNE	Precipitation No.	
New Oil	-	10.3	3.02	-	-
023-721	1	6.0	0.54	1.5	3.5
023-729	1	7.3	1.65	1.0	9.0
023-703	1	7.0	1.01	1.5	6.0
023-712	2	7.0	0.98	0.2	1.5
023-693*	-	5.2	0.25	2.5	13.0
*023-693 - 114,425 miles; others low mileage under 30,000. (Col. 6 with Flocculant)					

For the 1950 fleet a drain period of 2500 miles with a 500 mile leeway was established. Table II gives a few representative analyses of drain oil from this fleet and, for reference, the pH and total base-number of the unused oil. Of the four low-mileage cabs, note that 23-721 has had the alkaline reserve seriously depleted. A TBNE of approximately 0.3 is considered to be the low limit at which adequate protection against corrosive top-cylinder wear will be obtained. The other three low-mileage cabs have more than sufficient alkalinity to protect against acid corrosion. The high-mileage cab has had alkalinity

almost completely exhausted. However, at this mileage oil consumption has increased and it is estimated that 3 or 4 quarts of oil have been added as make-up, and that reasonably adequate protection is being obtained. Anticipated limited further service life and increased cost did not warrant corrective measures.

Columns 5 & 6 show two precipitation numbers. Column 5 is the standard ASTM method and column 6 is the same method with a flocculant added. The results of these two tests are interpreted as a measure of the detergent-dispersant properties of the oil when sampled. Note that in all cases the amount of sludge increases with the use of the flocculant. Experience indicates that all of these oils will satisfactorily hold sludge in suspension. When the results in column 6 are the same or less than those in column 5, the dispersant power of the oil has been exhausted and sludge will be deposited in the engine instead of being drained out.

Table III Analyses of drain oil 1953 fleet. (Oil drained 2500-3000 miles.)

Table III Lic. No.	Oil Added qts.	pH	TBNE	Precipitation No.	
New Oil	-	7.8	3.04	-	-
023-710	1	6.8	1.10	0.5	1.0
023-488	1	5.5	.37	1.5	17.0
023-221	1	5.2	.25	1.0	6.0
018-391	1	5.8	.43	1.0	4.0
023-708*	1	4.2	.12	11.0	8.0

*023-708 - oil drained 4297 miles; all new taxicabs less than 10,000 miles.
(Col. 6 with Flocculant)

Table III shows similar data for the 1953 fleet. Three samples are dangerously close to the low limit which was established for the base number. The first sample is satisfactory as regards alkalinity. The first four samples have adequate detergency to hold sludge in suspension and thus maintain a clean engine. As a result of the serious depletion of alkalinity, drain periods were reduced from 2,500 to 2,000 miles with a leeway of 300 miles. The analysis of the sample from 23-708, which was drained after 4,297 miles, shows that the alkaline reserve has been almost completely exhausted. This oil will no longer provide protection against corrosive wear. Furthermore, the precipitation number with the flocculant is less than that obtained without the flocculant, which indicates that the dispersant properties of this oil have also been exhausted, and that sludge was being deposited in the engine. This oil has been in use much too long and is presented as an example of how not to use additive oils. Note that in the 1953 fleet the drain period was reduced. Oil analyses indicated that a borderline case existed which did not justify a change to the extremely high additive series 2 oil, and that a slight reduction in the drain period would give the desired margin of safety.

Savings have amply justified the use of the high additive Supplement 1 oils in taxicab service. Fig. 8, showing relative maintenance cost in comparison with per cent increase in oil cost, demonstrates this. 1948 was taken as the base. Wartime cabs were still in service and it was not until near the end of the year that the entire fleet had been replaced. In 1949 maintenance costs calculated as cents per mile were reduced 14%, and in 1950 there was an increase of 3.6%. This fleet was replaced in the latter part of 1950, and during 1951 costs were reduced to 58% of the base value. In 1952 costs increased 17.2%. During 1949 a re-ringing program had already been instituted. With control of top-cylinder wear very little re-ringing was necessary during 1951. In 1952 a re-ring program was started and this accounts for the 17.2% increase. The average cost for 1949 and 1950 was 87.8%. For 1951 and 1952 it was 66.6% or a reduction of 21.2%. A further saving was the reduction in lost revenue time because of the cab being in the shop for engine over-haul. A third saving, intangible but important, was the improvement in driver morale.

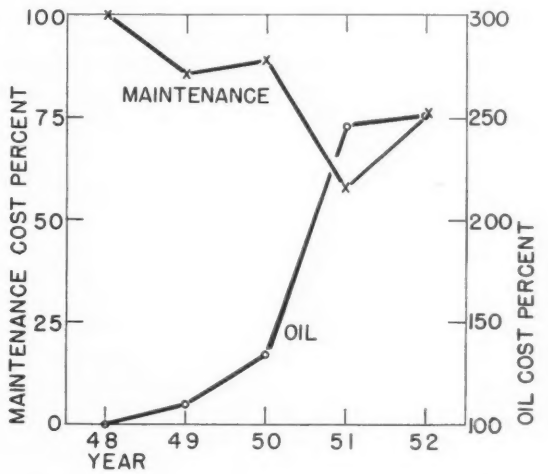

Fig. 8 Illustrates the saving that can be obtained through the use of the correct additive-content oil in a given engine and type of service. Relative maintenance costs calculated on cents per mile basis and oil costs on a cents per gallon basis.

CONCLUSIONS. Consideration of the factors presented above lead to the following conclusions regarding cylinder wear in taxicab service:

(1) Frictional wear of cylinder surfaces was a minor item and was not the limiting factor in engine life.

(2) Abrasive wear was, apparently, also of minor consequence.

(3) Corrosive wear, caused by the reaction of the acid products of combustion upon surfaces at temperatures below the dew point, was the major item causing cylinder wear.

(4) High operating temperatures alone do not effectively control top-cylinder wear. They do decrease the rate of wear and so prolong engine life.

The validity of this analysis is substantiated by the results obtained when a fleet of 1600 taxicabs

operating in New York City was supplied with an additive oil of Supplement 1 level. Top-cylinder wear was reduced to a degree where ring life was approximately doubled. When re-ringing became necessary, effective oil control could be re-established with the use of a minimum of new parts. Sludge deposition on or in critical parts of the engine was reduced or eliminated. Manual cleaning or solvent flushing was no longer necessary. Maintenance costs were reduced. Driver morale was improved and lost revenue time substantially decreased.

Book Reviews

AIR & OIL CYLINDERS IN INDUSTRIAL APPLICATIONS, by E. F. Heiser; The Industrial Publishing Co., 1240 Ontario St., Cleveland 13, Ohio, 1954, price $5.00.

Written by a practical man for practical men, **Air & Oil Cylinders in Industrial Applications** is devoted exclusively to fluid power cylinders. Mr. Heiser has attempted to answer some of the most frequently asked questions and problems he has encountered in 18 years of first-hand experience in working with plant people on cylinder design, selection and application. Twenty-one chapters stick close to the exact, detailed information that has proven most helpful. Chapter headings include "Cushion Limitations & Variations," with others devoted to Ordering Information, Operating Speed, Shock, Mounting Clearances, and Air Power-Oil Control; with a final chapter covering 12 capsule case histories of applications. A 10-page Appendix contains commonly used charts and tables, including specially prepared J.I.C. Hydraulic & Pneumatic component symbols.

Lubrication in the News

INCREASED LOAD-CARRYING CAPACITY. The load-carrying capacity of copper-lead and aluminum sleeve type bearings has been increased more than 100 percent by adding fine rouge (1-micron particles), a mild abrasive, in lubricating oil. H. G. Rylander & E. M. Wight, both associated with the University of Texas, report that research at the university on bearings of these materials lubricated with oil containing red rouge, molybdenum and corundum, has shown extremely important effects on friction and load-carrying capacity of bearings. Surface finish was found to be of prime importance. Rouge improved the finish under low load conditions before the high loads were successfully applied. Molybdenum sulfide was found to have good anti-seizing properties. This material, consisting of 39-micron particles, lowered the critical *ZN/P* ratio but did not lower the coefficient of friction in the region of film lubrication. It was most effective on copper-lead bearings, which had a relatively rough surface compared with the aluminum bearings. Load capacity was increased 60 percent with ½ gram per gallon of lubricating oil. No significant improvement had been noted in the use of the material in the softer Babbitt bearings. Small amounts of 6-micron corundum increased friction of clean oil about four times in aluminum and copper-lead bearings. Action of 22-micron corundum was to roughen the bearing surfaces. Wear under all loads was rapid when corundum particles were larger than oil film thickness. (Copies of this report, No. 54-F-11, may be obtained at 50c each from: Order Dept., ASME, 29 W. 39th St., N. Y. 18, N. Y.)

Lubrication Summarized

What Type of Packing or Gasketing Material Should I use with a Certain Hydraulic Fluid?, by H. C. Crosland, Chicago Rawhide Mfg. Co. ('54 National Conference on Industrial Hydraulics).

Halogenated hydrocarbons, such as the RPM fluids. Very rough on synthetic rubber packings. A low swell Buna N compound would be the best recommendation when temperature ranges from 0° F. to 250 F. Application should be checked because of possible volume increases of 30% to 40% with this type of fluid. Teflon or Kel-F, correctly designed, can be used with this and any of the other fluids. None appear to affect these two materials, and limitations of their use are physical limitations rather than effect of fluids on them.

Phosphate esters, such as Skydrol or Pydraul. Seals and packings made from butyl rubber or silicone are entirely satisfactory. Tests run as high as 212 F. indicate only nominal volume increase and durometer decrease with these two, properly compounded. Seals recommended for hydrocarbon base fluids are not usable when a switch is made to nonflammable phosphate esters.

Water glycol types, such as Hydrolube or Houghtosafe. Depending on the type of seal required and the temperature range involved, Buna N, neoprene, silicone and butyl all are satisfactory with this class of fluids, which was one of first to reach widespread use. These fluids give little trouble on all types of sealing applications in hydraulic systems, although different brands affect synthetic rubber compounds differently. At 212 F., a volume change of 1% to 25% can be found depending on make of fluid and type of synthetic rubber.

Silicone fluids. Good because of their excellent viscosity index. Sealing these fluids is more a design problem than a compounding one. Most synthetic rubbers will shrink when in contact with silicone fluids. The Buna N materials, low in acrylonitrile content, have a very low shrinkage when in contact with these fluids and appear to lend themselves best to this work. Also helpful are the silicone fluids having viscosities in excess of 100 centipoise because they have less effect on the rubber than those with low viscosities.

(Continued on p. 352)

JOURNAL & THRUST BEARING PRACTICES ON LARGE ROTATING ELECTRICAL MACHINES

by R. A. Baudry & B. B. Winer*

Rotating electrical machines cover a wide range of weight, speed, and sizes, from a fraction of a pound to over one million pounds, from a few revolutions per minute to several thousand revolutions per minute and from less than an inch in diameter to nearly 40 feet. The bearings supporting the rotors of these machines are of many types. Ball and roller bearings are extensively used on the smaller machines. Journal and thrust bearings having the load supported on a hydrodynamic film of oil are widely used on medium and large machines of ratings higher than 100 HP. This paper deals with these hydrodynamic thrust and journal bearings.

Among the large machines, the ones most often in the limelight are the generating units. These units are continuously increasing in rating and speed. The huge slow speed steam engines built fifty years ago have been replaced by high speed steam turbine generators having ratings up to 250,000 KW and a speed of 3600 RPM. The importance of such machines cannot be exaggerated. The present economic life of the world is dependent upon a continuous supply of electric energy. The failure of electric power in a community brings to an immediate stop the operation of many factories and affects the lives of all of us. It is obvious that the reliability of generating units is of utmost importance and that such major components as the bearings and lubricating system are required to give years of trouble-free operation.

Even larger physically than the turbine-generators are the huge modern vertical waterwheel generators. These units, running at speeds ranging from 80 to 600 RPM, have been made possible by the development of reliable thrust bearings. Such thrust bearings are now made as large as 10 feet in diameter and carry loads in excess of 4,000,000 lbs.

In parallel with the development of generators of electrical energy, there is an ever increasing use of electrical motors in industrial applications. Among the more spectacular applications are the motors for huge wind tunnels, mammoth irrigation pumps, and powerful steel mill drives. The majority of large motor applications, however, have a rating from 100 to 2000 HP and cover a speed range from a few RPM to 3600 RPM.

Large electrical equipment is required to have a long life and great reliability. It is the common practice for large generating units to operate for many months, even for years, without stopping. Many of the large generating units installed 30 or 40 years ago are still operating satisfactorily with their original bearings. In order to operate for such a length of time, bearings must be trouble-free, and oil leakage which might damage electrical windings must be eliminated. These rigorous service requirements have resulted in the development of bearings and seals which require a minimum of maintenance.

Journal and pivoted pad bearings in which the load is carried on a self-generated hydrodynamic oil film are particularly well suited for these applications since they can operate for years with practically no wear or loss of efficiency, and can be replaced or repaired when necessary with a minimum of effort.

Unit Loading & Geometry of Bearings. On waterwheel generators, a large part of the bearing load is due to the hydraulic forces of the waterwheel. On some special machines, magnetic forces can cause appreciable loads on bearings. In addition, there are belted or geared applications where the bearing load is affected by the external load. On most electrical machines, however, the bearing load is primarily due to the weight of the rotating parts.

As a result of this condition, the unit loading on electrical machine bearings is limited to a large extent by starting conditions where the bearing operates with solid or boundary friction. This is particularly true of reversing mill drives that frequently rotate at an almost negligible speed and turbine generator units that must be operated for many hours previous to starting at slow roll speeds as low as 3 RPM. Experience has shown that limit loadings up to 200 psi for journal bearings and up to 500 psi for pivoted pad thrust bearings are entirely satisfactory for most applications.

As a result of the constant vertical direction of the bearing load, a majority of the journal bearings used on electrical machines are of the partial type

*Westinghouse Electric Corp., Pittsburgh, Pa.

This paper was sponsored by the ASLE Technical Committee on Bearings & Bearing Lubrication, and presented at the ASLE 9th Annual Meeting, Cincinnati, April 6, 1954.

with an arc varying between 60 and 120 C., and with a large relief at the horizontal split. The only exceptions are bearings subjected to a large belt or magnetic pull which can act in any direction. In this case, the relief at the split may be omitted or a pivoted pad bearing may be used.

Lubrication. In order to perform satisfactorily, bearings must be provided with an adequate supply of oil. This supply of oil must be sufficient to establish a normal oil film and also to remove the frictional heat losses.

Because of its simplicity and reliability, the oil ring journal bearing is used on most of the medium speed rotating electrical machines. Its use has been extended to higher speeds by better proportioning of the heat flow path, forced ventilation, and water cooling of the bearing shell[1,2].

With increasing speed, the oil rings are supplemented by forced oil feed. For the high peripheral speeds used on turbine generators, oil rings are not practical and forced feed lubrication without oil rings is used. On vertical generators, the thrust bearings and guide bearings are usually operating in an oil bath. The friction losses are removed by means of water-cooling coils immersed in the oil.

The viscosity of the lubricating oil varies from 150 S.U.V. for high speed turbine generators, to 200 S.U.V. for medium speed oil ring bearings and to 300 S.U.V. for pivoted pad thrust bearings. Fig. 1 indicates typical lubrication specifications for electrical rotating machines.

Bearing Materials. The bearings used on electrical machines are lined with a low melting antifriction metal of the babbitt type. The principal purpose of the babbitt lining is to protect the journal surfaces from the scoring action of high spots resulting from machining variations or local high temperatures. After a relatively low number of starts, the high spots of the babbitt surface wear out and the fitting of the bearing is improved.

High tin base babbitt similar to ASTM Spec. B23, Alloy #2 is usually used for high speed turbine

	Horizontal Machines		Vertical Machines		
Oil Specifications	200 HP and up. Oil ring lubricated sleeve bearings; all speeds. Circulating oil system sleeve bearings; journal speed less than 2800 ft./min. See Notes 1 & 2	600 HP and up. Circulating oil system sleeve bearings; journal speed 2800 ft./min. and up	Kingsbury Thrust Bearings All ratings Speed less than 5000 ft./min. at bearing O. D.	Kingsbury Thrust Bearings All ratings Speed 5000 ft./min. and up at bearing O. D.	Guide Brgs. All Speeds All Ratings
Type of Oil	Light Machine Oil	Light Turbine Oil	Heavy Turbine Oil	Heavy-Medium Turbine Oil	When there are thrust bearings and vertical guide bearings on the same machine, use oil recommended for thrust bearing in guide bearing.
Flash Point, °F Min.	320	350	350	320	
Saybolt Viscosity	180-250	140-160	275-325	200-250	
Sec.-100°F	---	---	---		
Sec.-210°F					
Viscosity Index, Min.	50	50	50	50	
Carbon Residue, % Max.	.10	.10	.20	.15	
Neutralization No. Max.	.10	.10	.10	.10	
Pour Point, °F	See Note 3	See Note 3	See Note 3	See Note 3	
Rust & Oxidation Inhibitors	See Note 4	Yes	Yes	Yes	

Fig. 1 Oil Specification Chart. (Where special equipment or unusual operating conditions are involved, manufacturer should be consulted for lubrication specification.)

*When the lubrication system of the electrical equipment is part of the lubricating system of the turbine, follow turbine supplier's oil specifications.

Note 1. In general, for oil ring lubricated bearings the oil viscosity at the operating temperature should not be less than 70 SUV. Accordingly, where oil ring lubricated bearings operate continuously at temperatures above 158 F. (70 C.), a turbine oil having a viscosity of 300 SUV or more at 100 F. may be used. This may cause the bearing to operate with higher losses and at a slightly higher temperature, however, the highly refined turbine oils are more resistant to sludging, and the higher viscosity maintains the desired oil film at elevated temperatures. The operating temperature referred to above is the temperature as measured by a thermometer inserted in the oil ring inspection hole. This temperature is normally 5 to 15 C. lower than the temperature indicated by a thermometer imbedded in the bearing shell.

Note 2. On some applications, such as marine installations, temperature and losses permitting, heavier oils up to 800 SUV at 100 F. can be used in order to utilize the same oil employed in lubricating the main propulsion unit.

Note 3. As a guide for selecting the proper pour points the following table has been prepared:

Min. Ambient Temp. °F.	Max. Pour Point Temp. °F.
45 or above	35
32 or above	20
20 or above	0
0 or above	—10
Below 0° F.	Consult Manufacturer

On some applications rather than using an oil with a very low pour point it may be more practical to provide some heating means to assure the oil temperature will not fall below a certain minimum point.

Note 4. The majority of equipment in this classification will operate satisfactorily without inhibitors. Turbine-type oil containing inhibitors are recommended, however, on equipment operating under high speed, high temperature or high loading conditions. Oxidation inhibitors retard deterioration of the oil while the rust or water corrosion resistance inhibitors prevent corrosion of parts in contact with oil, particularly in the presence of water. In addition, some rust inhibitors tend to decrease starting friction, a desirable feature for highly loaded equipment. For applications where excessive foaming is a problem, special anti-foaming agents can be obtained.

generator and pivoted pad thrust bearing applications. For most applications, lead base babbitt, similar to ASTM Spec. B23, Alloy #7 with a low tin content of approximately 8% is used. During the war as a result of the shortage of tin, arsenic type lead base alloys with less than 1% tin were used and a few cases of babbitt corrosion resulted[3]. Such corrosion trouble has never been observed on the alloys which are now used.

One important factor in the manufacturing of a bearing is the bonding of the babbitt to the bearing shell. Most bearings shells are made of cast iron or cast steel. On cast iron bearings, it was common practice until a few years ago to provide cast anchor grooves to hold the babbitt. There was no attempt to obtain a metallurgical bond between the babbitt and shell. As a result, the babbitt was loose. This looseness increased with time on account of the appreciable temperature gradient and larger coefficient of thermal expansion of babbitt. After many years of service, as a result of the pulsating load usually present due to vibration of other factors, undesirable cracks would sometimes appear in the babbitt.

In recent years, new processes have been developed which permit cast iron to be tinned making it possible to obtain metallurgical bonding of the babbitt to the bearing shell. As a result, it has been possible to eliminate completely the anchor grooves and to produce babbitted bearings with a very sound bond. With the development of manufacturing processes such as centrifugal babbitting and water cooling of the shell to control the cooling rate of the babbitt, babbitt segregation has been eliminated and bonding even further improved. A valuable adjunct to our present manufacturing processes is the use of modern quality control methods. Fig. 2 shows the use of a fluorescent oil to detect flaws or porosity in the surface of the babbitt.

It is particularly important to have a sound bond between the babbitt and the bearing on water cooled bearings. In addition, attention must be given to the thermal gradient between the surface of the bearing and the surface of the cooling water. The large coefficient of thermal expansion of the babbitt, its low heat conductivity and tensile strength limit the permissible thermal gradient. In such designs, a large mass of babbitt around the cooling coils must be avoided in order to prevent excessive thermal stresses which can result in cracking of the babbitt. Fig. 3 illustrates a 20 x 40 water-cooled, oil ring lubricated, 720 RPM bearing that utilizes a copper lining in order to maintain a minimum thermal gradient between the babbitt and the cooling water[1]. High speed bearings of this particular type have been giving many years of trouble-free service. For lower speed water-cooled bearings the copper sleeve is not necessary as shown in Fig. 10.

Fig. 2 Fluorescent oil inspection of Babbitt surfaces to inspect flaws.

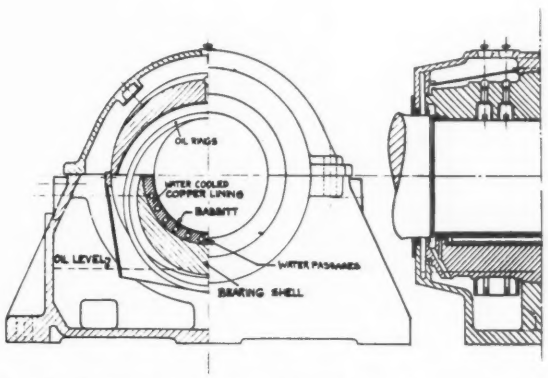

Fig. 3 Oil ring water cooled journal bearing, 20 in. dia., 720 RPM.

Recently, it was discovered that steel slabs used to produce large thrust bearing pads contained an appreciable amount of hydrogen in solution. The relative volume of hydrogen contained in the steel was found to be as high as 75% of the volume of the steel, when referred to atmospheric pressure at 0° C. Investigation revealed that this hydrogen was diffusing slowly through the steel, but not through the babbitt. Therefore, when the hydrogen reached the interface between the steel pad and babbitt, it would accumulate causing high pressures. This pressure forced the babbitt away from the steel so that blisters were formed which resulted in bearing difficulties[4]. This condition was corrected by suitable heat treatment of the steel before babbitting. Since the rate of diffusivity of hydrogen is inversely proportional to the square of the thickness of a steel slab, hydrogen has been found to be a problem only in the very large bearings.

Method of Supporting Bearings. Many methods of supporting the bearing shell in the pedestal or bracket are used. On the smaller size of machines provided with a structure that can be aligned accurately and with relatively stiff shafts, fixed sleeve type bearings are successfully used as shown in Fig. 4a. It is an economical design because of the simple construction and the fact that the bearing

shell can be made of minimum thickness. Furthermore, the large contact area between the shell and supporting bracket results in a low resistance to the flow of heat and a low operating temperature. On machines provided with long flexible shafts, or where the bearing shells are supported in pedestals mounted on a foundation subject to distortion, it is the normal practice to use self-aligning bearings. Several types of self-aligning bearing seats are used. The narrow seat shown in Fig. 4b is most widely used. It is employed on most medium speed machines; it is economical to produce, and can follow changes of shaft position very accurately since the friction forces are very small. The spherical seat shown in Fig. 4c is used on high speed machines. This type of seat has higher friction forces and cannot realign itself with the shaft as well as the narrow seat. However, it has a major advantage in that it can be operated self-cooled at higher speeds since the wide spherical seat offers a very low resistance path to the flow of heat. In addition the wide seat is less sensitive to wear caused by vibration.

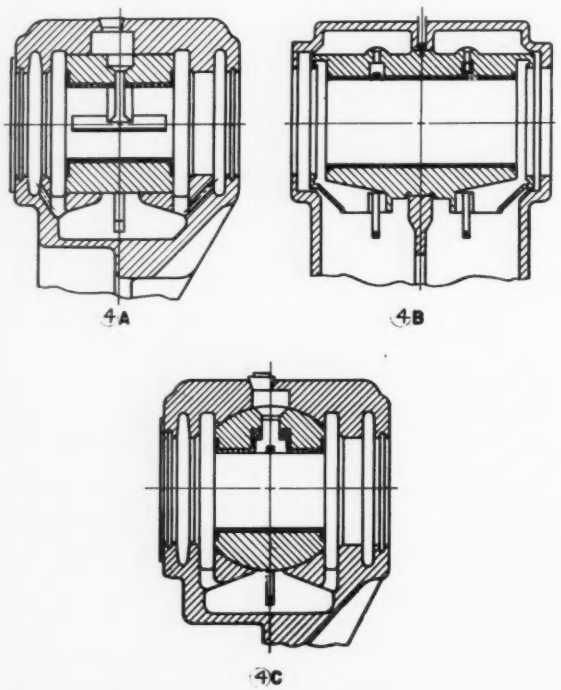

Fig. 4 (A) Fixed bearing; (B) Narrow seat self-aligning bearing; (C) Spherical seat self-aligning bearing.

Oil Seals. Bearings used on electrical machines must be provided with very effective seals. Even a small quantity of oil mist allowed to escape from the bearing will, after a period of years, cover all the windings of the machine with a thin layer of oil. Oil cannot be tolerated on windings because of its damaging effect to the insulation and its adverse effect on electrical creepage distances. In addition, the air used to ventilate the windings contains an appreciable amount of dust which collects on the oil deposited on the windings. After a period of time, this dust forms a thick layer of dirt packed solidly on the windings and interferes with the ventilation.

After years of experience, very satisfactory seals have been developed. Their design depends on the speed of the machine and location of the bearing with respect to the ventilating system. In addition to labyrinth seals of various types, effective seal systems are used that control the air pressure at the shaft seals in order to prevent the suction of oil-laden air from the bearing housing.

On low speed machines, felt seals are occasionally used, however, on high speed machines such seals are objectionable because the friction forces and heat generated at the point of contact are sufficient to cause severe unbalance forces and resulting vibrations. In addition, excessive wear of the felt seal would occur. For these reasons, on medium and high speed machines thin labyrinth seals having a radial clearance slightly larger than that of the bearing are used in order to prevent any possible contact with the shaft.

Fig. 5 illustrates a typical bearing and its housing. Due to the fan action of the rotor, shaft flange, and collector rings, there is usually a difference of pressure between the two ends of the bearing which attempts to make air flow through the bearing. This air carries with it some of the oil mist always present inside the bearing. The first step to reduce the oil leakage is to keep to a minimum the oil mist inside the bearing housing. Referring to Fig. 5 note the labyrinth seals at the bearing ends, the grooves near the ends of the bearing and the recesses to prevent oil from leaking along split[5]. The provision of a barrier preventing air flow between the bearing and its housing also helps to keep air circulation to a minimum. When the air pressure differential between the two ends of the bearing housing is appreciable, it is good practice to provide a pressure-equalizing passage connecting both ends of the

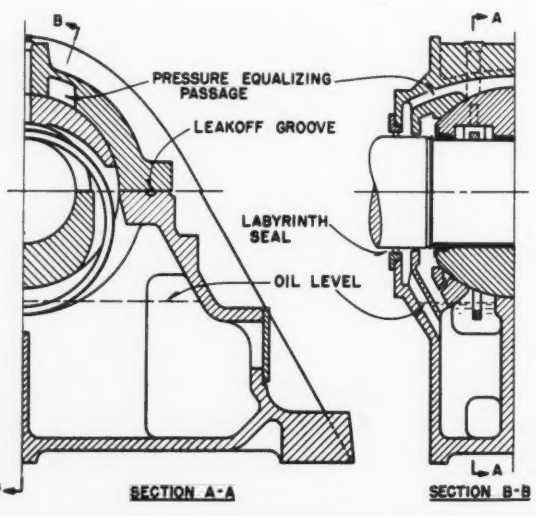

Fig. 5 Spherical seat bearing and pedestal.

housing. This chamber establishes the same pressure at both ends of the housing, thus preventing the flow of oil-laden air to the outside of the bearing housing. This type of seal has been very successful in medium speed machines.

On high speed machines provided with forced feed oil circulation, there is an oil drain connected between the bearing housing and a remote oil reservoir. The air pressure at the ends of the bearing housing is sometimes lower than that at the oil reservoir. Therefore, air laden with oil mist can be drawn through the drain pipe and bearing housing thus causing undesirable oil leakage. This condition can be corrected by means of a pressure seal as shown schematically in Fig. 6, which establishes at the bearing seals a pressure higher than that in the oil reservoir. On electric machines, it is usually possible to tap air from the high pressure zone of the ventilating system.

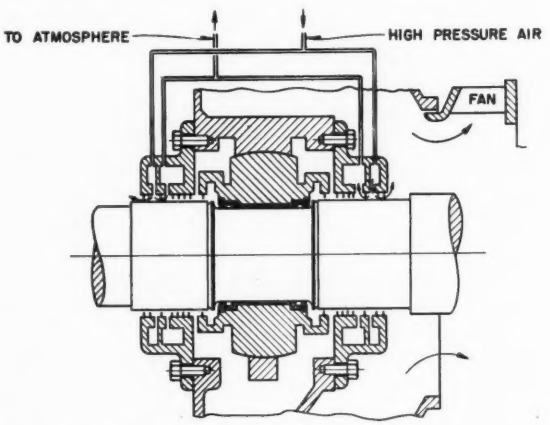

Fig. 6 Bearing with positive oil vapor seal.

In cases where the above high pressure zone is not available or accessible, the same effect can be obtained by installing a suction fan on the drain system. This fan establishes in the bearing housing a pressure lower than that at the seals and forces the air to flow into the housing. As a result oil vapor is prevented from flowing out of the ends of the bearing housing. This scheme can be used only when the air around the bearing is reasonably clean; otherwise, there would be a possibility of introducing an appreciable amount of undesirable dust into the lubricating system.

In bearings provided with forced oil circulation, in addition to the above features, particular attention must be given to the design of the drain, in order to avoid double traps or other obstructions to the free flow of oil. Such obstructions would cause foam to build up in the bearing housing and leak out along the shaft. Under all circumstances, there must be a clear path above the oil level and the inside bore of the drain pipe to permit free circulation of air and foam to the drain connection. Particular attention must be paid to the design of the oil drain connection to insure the free flow of foam to the oil tank, where sufficient surface must be provided for settling the foam and deaerating the oil.

Calculation of Bearing Performance. The satisfactory operation of a bearing is usually indicated by continuous operation at a stable temperature. It is also usually desired to have a low operating temperature; however, on many applications at high speed or in hot surroundings high bearing temperature operation is inherent and is entirely satisfactory with an adequate design.

The calculation of the operating temperature of a bearing necessitates the accurate knowledge of the friction losses and of the characteristics of the various components of the heat flow path from the bearing surface to the cooling medium. Since these characteristics are not always known accurately, the operating temperature of a bearing is sometimes difficult to predict.

Guide bearings and thrust bearings for vertical machines are usually operating in an oil bath which is cooled by water cooling coils immersed in the oil. It is usual practice to design the cooling coil for a temperature rise of the oil of 20 C. above the cooling water.

High speed journal bearings are usually flood-lubricated and supplied with 0.5 to 1 gpm of oil per KW of calculated losses, which results in a temperature rise of the oil from 10 C. to 20 C. External oil coolers cooled by water or air are provided to keep the oil inlet temperature at 45 C. or less.

Most of the medium speed electric machines are provided with oil ring lubricated bearings. The prediction of the temperature rise of this type of bearing is more difficult since it involves the flow of heat through many complicated paths. Furthermore, the dissipation of heat at the surface of the bearing pedestal varies between wide limits depending on the temperature and velocity of the surrounding air. However, surprisingly accurate calculations have been made which have led to a better understanding of the heat flow through this type of bearing and an improvement in its performance[1].

The friction losses of bearings can be calculated according to methods originally developed by Reynolds and further amplified by Kingsbury[6], Sommerfeld [9,10,11] and many others. Convenient tables have been published simplifying bearing calculations [7,8,13].

On electric machines, the bearing pressure as mentioned earlier in this paper is determined by experience as the load which can be safely carried during the starting period or under slow roll when boundary friction exists. The oil film thickness at normal speed usually is very ample and its calculation is of secondary importance. It is useful, however, for determining the oil flow conditions and the spring constant of the oil film, or the alignment of high speed machines.

The simplest bearing is the guide bearing where the oil film is assumed to be uniform. The friction or shear force at the surface of such a bearing is:

$$F = 1.45 \times 10^{-7} \; Z \; AU/h$$

The characteristics of a pivoted thrust bear-

ing are given in Fig. 7 as a funtion of the ratio h_i/h_o of the oil film thickness at the inlet and outlet. It should be noted that the coefficient of friction and oil film thickness do not vary appreciably for values of h_i/h_o between 2 and 4 where the minimum coefficient of friction and maximum oil film thickness occur. These characteristics are for an ideal thrust bearing pad having the pivot offset the optimum value from the center of the pad, having an infinite width, perfectly plane surfaces and constant oil viscosity throughout the oil film. According to these assumptions a pivoted thrust bearing pad with a central pivot would not sustain an oil film. However, many years of experience have shown that a centrally pivoted pad bearing operates very satisfactorily. This paradox is explained by the variation of the oil viscosity[11] and the slight curvature of the pad resulting from manufacturing variation and temperature distortions[14].

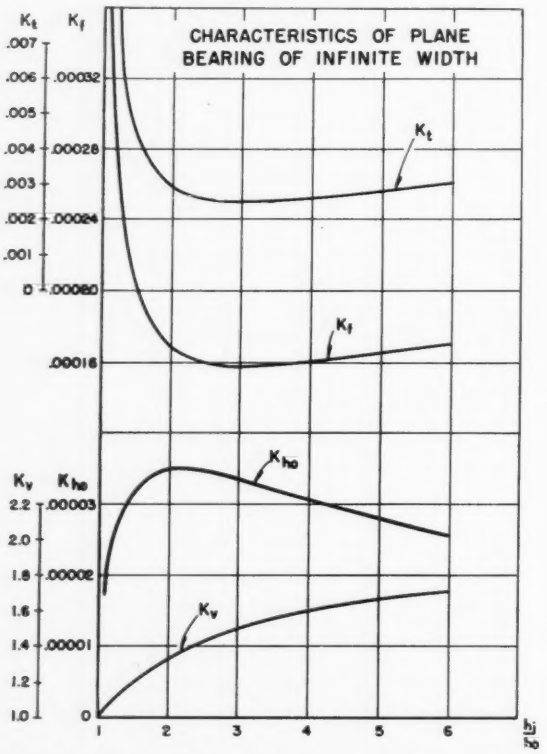

Fig. 7 Characteristics of plane bearing of infinite width; coefficient of friction = f = K_f (D/b)$^{1/2}$ (ZN/Pr)$^{1/2}$; minimum oil film thickness = h_o = K_{ho} b (D/b)$^{1/2}$ (ZN/Pr)$^{1/2}$; temperature rise of lubricant = T. = K_T P_r.

Experience has shown that a centrally loaded pivoted pad will maintain a tapered oil film having ratio of inlet to outlet oil film thickness, h_i/h_o, between 2 and 4, that the coefficient of friction and oil film thickness can be determined with sufficient accuracy by using the following average values for the coefficient of friction and minimum oil film thickness[9].

$$f=0.00016\ (D/b)^{1/2}\ (ZN/P_r)^{1/2}$$
$$h_m=0.00003456\ (D/b)^{1/2}\ (ZN/P_r)^{1/2}$$

These values are for an infinitely wide pad.

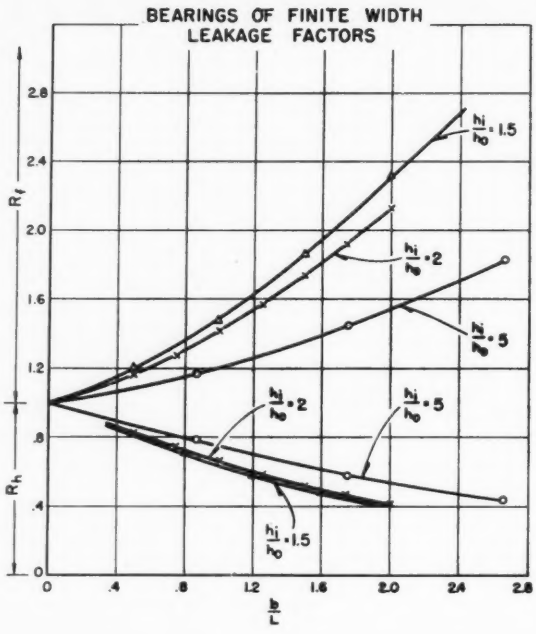

Fig. 8 Bearings of finite width—leakage factors; coefficient of friction = f = R_f K_f (ZN/Pr)$^{1/2}$ (D/b)$^{1/2}$; minimum oil film thickness = ho = R_h K_h b (D/b)$^{1/2}$ (ZN/Pr)$^{1/2}$.

For a pad of finite width the coefficient of friction and oil film thickness should be corrected by the factors given in Fig. 8.

The coefficient of friction and oil film thickness of journal bearing can be calculated as a function of the Sommerfeld number:

$$S=(C/R)^2\ (ZN/P_j)$$

Most of the journal bearings used on electric machines have a clearance ratio varying between .001 and .002. The ratio of .001 to .0015 bearing being used for low and medium speed machines, and .002 for high speed bearings. The bearings are provided with wide reliefs at the split which results in symmetrical bearings having an angle of 90°.

The characteristics of a 90° journal bearing are given in Fig. 9 as a function of $1/S^{1/2}$ and of the eccentricity. This method of representation indicates that between the wide range of eccentricities usually employed the coefficient of friction and oil film thickness are nearly independent of the clearance ratio and can be calculated with reasonable accuracy by taking

$$f=.0002\ (ZN/P_j)^{1/2}$$
$$h_m=0.000025\ d\ (ZN/P_j)^{1/2}$$

These values are for bearings of infinite width. The effect of finite width can be determined by using the same factors as given in Fig. 8 for plane bearings.

Allowance must also be made for the friction losses in the non-active parts of the bearing, such as reliefs, upper part, thrust shoulder, etc. Results of tests shown on Fig. 11 on some medium speed bearings indicate that the coefficient of friction can be taken for this type of bearing as

$$f=0.00035\ (ZN/P_j)^{1/2}$$

This method of giving the characteristic of bearings as a function of the characteristic number ZN/P has been used by many authors[10,19,20,21,22]. It is very convenient from the design standpoint, and has been shown by experience to be sufficiently accurate for most engineering applications.

The friction in slow speed lightly loaded bearings or high speed bearings having a large oil film thickness can be calculated with sufficient accuracy by assuming a uniform oil film thickness as for a guide bearing, which is the method used originally by Petroff. However, when designing bearings for special applications as for loading beyond normal practice, it is good practice to determine the characteristics more accurately according to the Sommerfeld number as shown on Fig. 9.

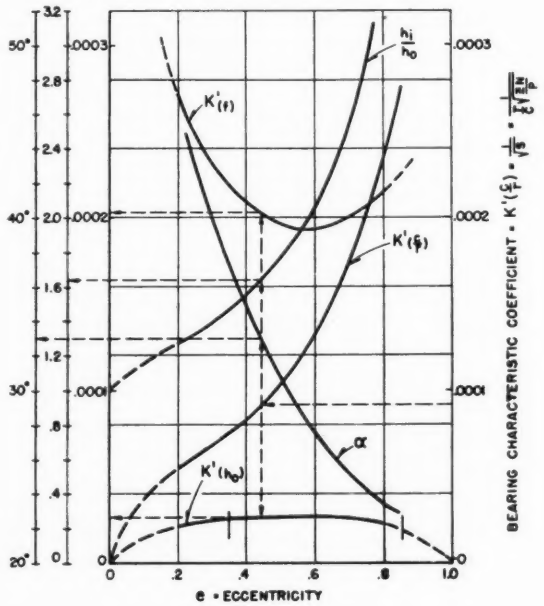

Fig. 9 Characteristics of 90° central partial bearings of infinite width. Near optimum condition. .35 <e <.85, coefficient of friction, f = .00027 $(ZN/P)^{1/2}$; minimum oil film thickness, h_m = .000025d $(ZN/P)^{1/2}$.

Requirements of Bearing Surfaces. In order to perform satisfactorily under high loading during the starting period and before an oil film is established, the bearing surfaces must be finished to high standards of accuracy and finish. The distortion of the journal and bearing shell due to heat or load must be kept to a minimum. Machining variations must be such as not to interfere with the normal formation of the oil film or result in localized high pressure loading during the starting period.

Bearings with flexible shells or shafts usually must be scraped under load to obtain proper operating conditions. Bearings in which the deflections due to load are small can be machined accurately and do not require any fitting under load.

It is necessary to machine journals to a true round surface; otherwise load pulsations and undesirable vibrations can result. Experience has shown that an elliptical or three noded journal can cause excessive vibration if the natural frequency of the shaft corresponds to the product of the number of nodes times the running speed.

Bearings used on electrical machines must start from rest at the maximum unit pressure and operate under solid or boundary friction. As a result, a good journal surface finish is essential. Experience and tests indicate that a roughness of 20 to 30 micro-inches is satisfactory; however, journals are usually machined so that the surface roughness does not exceed 16 micro-inches. Thrust bearing runners which are lapped under ideal conditions are polished to a surface roughness not in excess of 10 micro-inches[12]. In order to maintain the proper smoothness through years of operation, bearings must be well protected from corrosion and dirt in the oil.

The breakaway coefficient of friction of a journal or thrust bearing varies between .2 and .4. For some applications such as synchronous condensers, the available electrical torque is not sufficient to overcome the solid friction. It is then necessary to establish an oil film prior to starting by introducing oil between the bearing surfaces by means of a high pressure pump. An extremely small coefficient of friction can be obtained by this method[15,16].

Bearing Currents. In electrical machines, flux pulsations linking the electrical circuit formed by the shaft, bearings, pedestals and bedplate can cause an e.m.f. and large currents[18]. Experience has

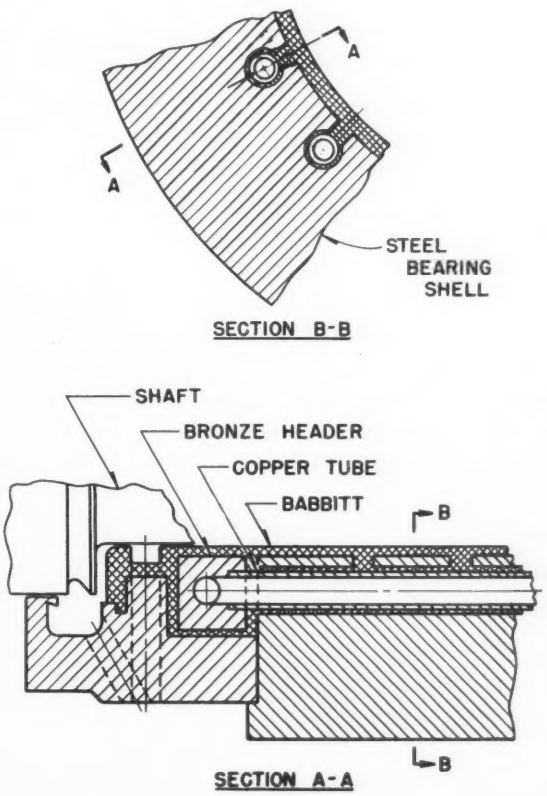

Fig. 10 Water cooled bearing.

shown that this current may result in considerable damage to the bearings. On units susceptible to this type of difficulty, it is accordingly good practice to electrically insulate the bearings to break the circuit. Sometimes the two bearings of the machine are insulated to facilitate checking the insulation. In some cases, insulation is placed under a pedestal or between a bracket and frame, in other cases it is put directly around the bearing itself. It is important to keep this insulation in good condition and to insure that it is not accidentally short circuited by thermometers or oil piping.

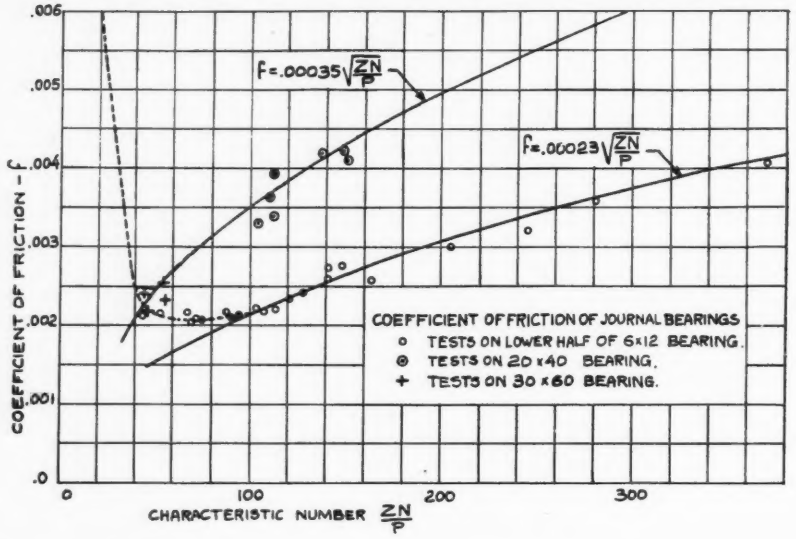

Fig. 11 Coefficient of friction of journal bearings.

Conclusion. Electrical machines have the common requirement of being able to operate continuously for long periods of time with practically no wear or loss of efficiency. An important factor in the art of building such machines has been the corresponding progress in the design of bearings based on improved manufacturing methods and judicious application of theory. Today we have journal bearings with peripheral speeds of 18,000 feet per minute and thrust bearings that carry loads of 2000 tons. The life of these bearings is measured in decades and they are built to outlast the equipment on which they are installed.

APPENDIX

Method of using bearing performance curves to calculate the friction factor and the minimum oil film thickness for a 90° central partial journal bearing.

Example: A 4" x 8" 90° central partial journal bearing is operating at 1200 RPM, the bearing load is 1800 pounds and the oil film temperature is 55 C. The oil is SAE 10. Determine coefficient of friction and minimum oil film thickness. d = bearing diameter = 4"; L = bearing length = 8"; Z = oil viscosity, 15.4 cp for SAE 10 at 55 c.; N = 1200 RPM.

These values give a unit pressure projected on the bearing diameter of 150 psi, $ZN/P = 123.2$, and a bearing characteristic number

$$K'\,(C/R) = \frac{1}{R/C\,(ZN/P_j)^{1/2}} = .00009$$

Entering Fig. 9 with this value of the bearing characteristic number, read $h_i/h_o = 1.63$, $K'\,(f) = .000203$ and $K'\,(h_o) = .000025$.

These coefficients are for bearings of infinite width and must be corrected for a bearing of finite width.

To accomplish this, enter Fig. 8 with $b/L = 0.785$ and $h_i/h_o = 1.63$. Read $R_f = 1.34$ and $R_h = 0.73$.

Using these values, calculate $f = R_f\,K'\,(f)\,(ZN/P_j)^{1/2} = .00302$ and

$$h_o = R_h\,K'\,(ho)d\,(ZN/P_j)^{1/2} = .000810.$$

Note that by using the approximate formulae, $f = .0002\,(ZN/P)^{1/2}$, and $h_m = .000027d\,(ZN/P)^{1/2}$, given on page 9 and the correction factors obtained from Fig. 8, the coefficient of friction becomes $f = .00298$ and the minimum oil film thickness becomes $h_m = .00081$. These values are in good agreement with those already obtained.

Fig. 11 gives the coefficient of friction as obtained in actual tests. The lower curve shows the coefficient of friction for the lower half of a 6x12 bearing, for which $f = .00023\,(ZN/P)^{1/2}$. The upper curve includes losses in the non-active parts of the bearing and is the one which should be used to determine the bearing friction losses.

TERMINOLOGY

A	bearing area (sq. in.)
b	width of the bearing surface parallel to the direction of motion (in.)
C	radial bearing clearance (in.)
d	journal bearing diameter (in.)
D	mean diameter of a plane bearing (in.)
e	eccentricity of journal bearing (in.)
f	coefficient of friction
F	friction force (lb.)
h_i	oil film thickness at inlet (in.)
h_m	minimum oil film thickness (in.)
h_o	oil film thickness at outlet (in.)
K_f	coefficient of friction factor for plane bearing of infinite width
K_{ho}	minimum oil film thickness factor at outlet for a plane bearing of infinite width
K_t	temperature rise factor for a plane bearing of infinite width
K_q	oil flow factor for a plane bearing of infinite width
$K(_f)$	coefficient of friction factor for a journal bearing of infinite width
$K(_{ho})$	minimum oil film thickness at outlet for a journal bearing of infinite width
$K(_{c/r})$	clearance ratio factor for a journal bearing of infinite width
L	width of the bearing surface perpendicular to the direction of motion (in.)
N	speed of the bearing (RPM)
P_j	unit bearing pressure proected on diameter (lb/in²)
P_r	unit bearing pressure for a rectangular plane bearing (lb/in²)
R	radius of journal bearing or mean radius of thrust shoe
R_f	leakage factor for coefficient of friction (in.)
R_h	leakage factor for minimum oil film thickness
S	Sommerfeld number
ΔT	temperature rise of the oil through the film (°F.)

U velocity of the moving surface (in/sec.)
V flow of oil in an infinitely wide bearing (in^3/sec.)
Z absolute viscosity (CP)
α angle between location of minimum oil film thickness and bearing load (degrees)
Viscosity (CP) = (6.9 x 10^6) Viscosity (Reyns)

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New Products

HI-PRESSURE XACTO METERS. Two new hi-pressure Xacto meters have been designed to give positive displacement accuracy on industrial liquid measuring jobs which require high-pressure applications. Known as 470-15-4 (rated at 15 gpm, with a working pressure up to 500 psi), and 470-50-4 (rated at 50 gpm, with working pressures up to 500 psi), the smaller model is furnished with 1¼" screwed connections, and the larger meter has 1½" flanged connections. Working parts of these meters are stainless steel, hard carbon bearings, and Teflon pistons, and may be equipped with straight-reading, totalizing, or ticket-printing dials. Both models are designed for either top or bottom mounting. For further details, write: Bowser, Inc. (LE10/6), 1358 Creighton Ave., Ft. Wayne 2, Ind.

HOUGHTO-CLEAN 402-403. A new combination "cold" cleaner, Houghto-Clean 402-403, has been developed to replace highly volatile solvent cleaners in still tanks on the production line. Mixing readily and easily in economical proportions with cold water, other advantages include: thorough removal of most generally encountered shop soil, protection of freshly cleaned surfaces from rusting between operations or in temporary indoor storage, elimination of heating system costs, and improved working conditions by the elimination of steamy atmospheric conditions associated with hot alkaline cleaning methods. For complete details, write: E. F. Houghton (LE10/6), 303 W. Lehigh Ave., Phila. 33, Pa.

SAFETY PANEL. A safety panel to protect unattended or remote-controlled engines from burned bearings, frozen pistons, or other costly failures has been introduced consisting of an oil pressure gauge and water temperature gauge in combination with an engine shut-off switch, an ammeter, stop switch and reset, and 14 amp fuse. (Designed for engines with battery ignition, other panels are also available for magneto ignited and diesel engines.) Should the oil pressure of the engine drop too low, or should the engine temperature rise to a dangerous level, the panel automatically shuts off the engine. The minimum oil pressure or maximum engine temperature at which it is desired to shut off the engine is pre-determined by the position of "stop" on the face of the gauges. The electrical contact points on the gauges are of silver to prevent corrosion and to assure clean, positive contacts. The panel case is of heavy gauge steel with a simulated hammered aluminum finish; it measures 12" long, 5⅛" wide, and is 2⅝" from front to back. For complete details, write: Stewart-Warner

(Continued on p. 352)

MEASUREMENT OF LUBRICANT THICKNESS ON POWDER-METALLURGY DIES

by N. P. Pinto*

The air gauge has been found to be a useful method for measuring the thicknesses of lubricant applied to the bores of dies used for making powder-metal compacts. Readings are accurate and reproducible within .00005". Besides affording a control for the development of satisfactory lubricant application technique, the information gained on die bore diameter has proved useful for analyzing causes of early die failure.

Lubrication of dies used in the production of powder-metal compacts may be effected by the addition of lubricant to the powder charged into the die or by applying lubricant to the die walls. For the latter method, the thickness of the lubricant coating is critical and must be controlled accurately over the whole working area of the die. An accurate and practical method for measuring lubricant thickness is necessary to develop acceptable procedures for applying lubricant and for controlling these procedures during production. Considerable effort has therefore been directed toward evaluating various methods of measuring lubricant film thickness.

It is the purpose of this paper to discuss the development and use of the air gauge method of measurement of lubricant coating thicknesses and to consider its advantages and disadvantages. The method is applied specifically to measuring lubricant films produced by spraying suspensions of colloidal graphite and/or molybdenum disulfide in volatile carriers into the bore of the die. However, respective merits of different types of lubricants and the effects of lubricant on product quality or die life are outside the scope of this paper and are not discussed.

Powder-metallurgy dies may be separated into two general groups, solid dies and come-apart dies. The former are far more extensively used. They may either be solid steel, with suitably machined die cavities, or composites with several sections and inserts. The come-apart die differs principally in that it is disassembled after each pressing for removal of the compact and must be constructed accordingly. The come-apart die finds wide application when the compacts of a particular metal or alloy are characteristically fragile, as is the case of tungsten. After the compact is pressed, both the compact and the die segments are ejected from the back-up die and the segments are stripped away from the compact.

The studies described herein were initiated to measure the lubricant thickness on a round, come-apart die consisting of cylindrical segments held in a closely fitting, solid back-up die. While it would thus be possible to consider methods for measuring lubricant coatings on external surfaces (as well as those for internal surfaces) since the die could be disassembled after lubrication for coating measurement, methods which measure only external coatings were discarded. Such methods would be limited to the evaluation of thickness applied by a given process. Since the lubricant coating would be destroyed at the parting line between die segments when the die was disassembled for coating measurement, methods limited to external measurements could not be applied to a die which would be returned to service and could not be considered a production tool for process control. Further, the external method could not be used to study such factors as the effects of a measured coating on compact characteristics nor the amount of the applied load transmitted to the die by die wall friction. On the other hand, a measurement of internally applied coating would evaluate the thickness directly without disassembling the die and results could be applied directly to plant operations. Only the latter method would permit a routine measurement without interrupting operations.

METHODS OF MEASUREMENT CONSIDERED. Including the air gauge, the methods which have been studied are: (1) Elcometer method of measuring coating magnetically, (2) Branson gauge, similar thereto, (3) Superficial Rockwell measurement of coated die hardness, convertible to coating thickness, (4) micrometer measurement of bore dimensions, (5) dial, or (6) air gauge measurement of the contour of a segment of the die bore, Fig. 1, (7) determination of coating volume by weighing the sprayed bore, and (8) determination of volume of a coating which had been applied and then removed by washing, etc.

The magnetic methods, (1) and (2), are not suitable since it is virtually impossible to apply the

*Sylvania Electric Products, Inc., Atomic Energy Div., Hicksville, L. I., N. Y.

measuring instrument to identical points on the die surface before and after lubrication. Further, the reading is affected seriously by variations in thickness of oxide coatings. The hardness measurement, (3), has been developed and evaluated at the Bayside Laboratories of Sylvania Electric Products, Inc.[1] The Superficial Rockwell Hardners Tester is used to apply a minor load and then a major load to the lubricated surface; the hardness number obtained is converted to coating thickness using a calibration made previously between the hardness tester and a shadow gauge. The hardness method is more simple and may be used at elevated temperatures; the air gauge method may be used on internally coated solid dies or on come-apart dies without disassembling and may be used as a production control tool.

The measurement of bore sizes micrometrically, (4), must be done with extreme care and precision, and duplication of readings is most difficult. The method measures the thickness of a coating compressed by the force of the micrometer rather than that of the porous, as-deposited coating.

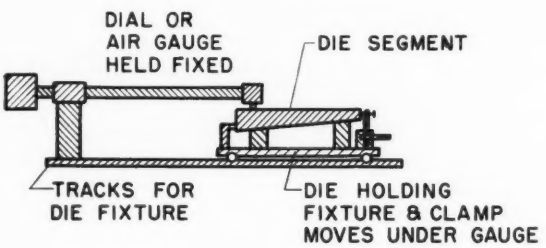

Fig. 1 System for measuring bore contour.

Determination of bore contour, (5) or (6), is considered to be the most practical and useful for the measurement of coating thickness for either internal or external surfaces. One arrangement uses a jig, as shown in Fig. 1, in which an air gauge is held fixed while the surface (of a die segment) is moved under it. An accurate surface contour is obtained by plotting the gauge reading against location on the surface, after which the die segment is removed from the fixture, lubricated, and carefully repositioned in the fixture and the contour is remeasured. The difference between original and final contour is, of course, the coating thickness. For reasons given, this method as applied to a die segment would not be suitable as a production control tool, nor would it be suitable for the measurement of bores smaller than about 1.5″ for mechanical gauging and 0.2″ for air gauging.

Methods (8) and (9) would evaluate a lubrication method only, since they would yield an average coating thickness; moreover, the information would be of limited importance and could not be correlated with production data.

None of these methods, except (4), (5), and (6), would be adaptable to the measurement of bores of assembled dies; this is an important consideration. If the adaptability of the accepted method to usage with hot dies is considered important, the practical value of all seven methods is reduced further.

Several available bore measuring systems employ a diamond or metal point which is moved along the bore. Depending on the radius of the indicating point, plowing the soft lubricant could become an important source of error and give false readings.

In view of these practical and theoretical considerations, together with considerable experimental work, it was decided to concentrate efforts on evaluating and refining the air gauge method, applying it to measurements of the assembled die rather than to a die segment.

METHOD OF AIR GAUGING. The air gauge system is shown schematically in Fig. 2. The air gauge is based on a measurement of the flow of an air stream which passes from the flowmeter through the handle of the gauge into the mandrel, where it splits and flows out two radial holes. After leaving the radial holes, the air stream passes between the outside diameter of the mandrel and the bore of the die. Since the air is supplied at constant pressure, the greater the clearance between mandrel and die, the greater will be the air velocity. By measuring bores of known diameter, the air flow readings can be calibrated to read true bore diameters.

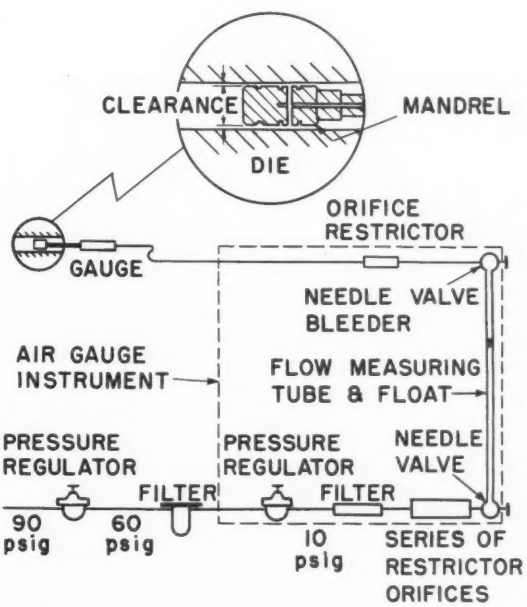

Fig. 2 Schematic diagram of air gauging system.

The thickness of a coating on the die bore is determined by measuring one element (line) of the bore (a) before and (b) after being lubricated. The difference between the two readings is twice the average coating thickness. In order to be certain that the same element is measured before and after lubrication, the die is placed in a positioning fixture (Fig. 3). It is held solidly by an end clamp forcing the die against a stop and by a side clamp

positioning it in V-ways. The die is indexed (azimuth) with a reference mark on the die aligned with a reference point on the fixture.

With the gauge mandrel in the bore of the die, the hexagonal gauge handle is moved on ways, indexing the position (depth) of the measuring point of the mandrel within the bore. The exact location of the diameter being measured is read on the scale on the fixture.

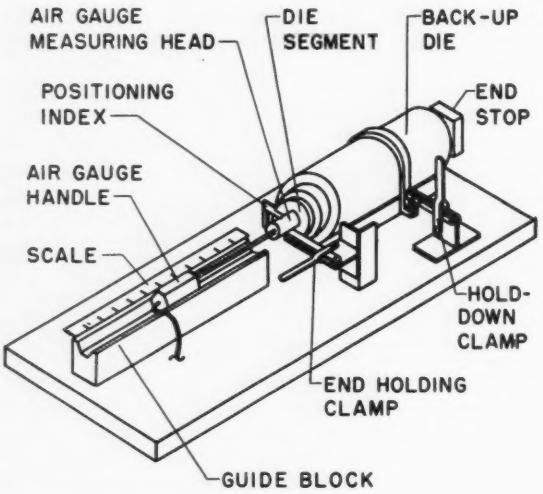

Fig. 3 Fixture for measuring die bores.

In operation, the die is cleaned and clamped in the fixture; the bore is measured at various points along its length, and the contour of one element of the bore is plotted, as in Fig. 4. Then the gauge is withdrawn and its hexagonal handle is rotated 60 degrees and a second contour is determined. This procedure is repeated for the third contour 120 degrees away from the first diameter. The die is removed from the fixture, lubricated, and then remeasured along the same elements as above. When a die is characterized every ¼ inch, a total of 200-300 readings may be required, depending on the die length.

ACCURACY & REPRODUCIBILITY. Considerable experience with the air gauge has established with reasonable certainty the accuracy and reproducibility of the method. In the 1000/1 amplification employed, diameters may be read accurately to ± .00005". Reproducibility is also within ± .00005".

TYPICAL RESULTS. The dimension measured is bore diameter but is generally converted and plotted as the average bore radius, so that the difference between the two contours (before and after lubrication) is the average coating thickness. A typical plot is shown in Fig. 4, each contour being magnified about 175 times. This plot characterizes one particular application of lubricant and shows that the operator applied a coating of about .0004 inch average thickness, that the coating was slightly thinner near the ends than at the center of the die, and that at no spot was the coating greater than .0005" nor less than .0002" thick.

This system provides a comparison between the coatings applied from run to run and also from operator to operator. Dies are lubricated by spraying suspensions of colloidal graphite and/or molybdenum disulfide in volatile carriers into the bore of the die. It has been found, for example, that different operators using the same spray gun settings will apply coatings which are consistent for a given operator but which vary significantly with the operator. Thus, an operator applying a consistently light coating may compensate by adjusting the equipment.

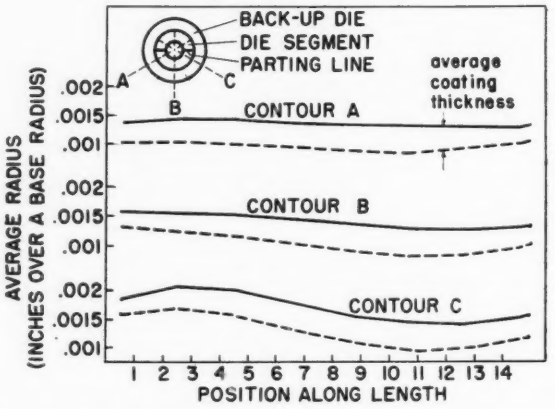

Fig. 4 Contour of the bore of a die before and after lubrication. Radius plotted in thousandths of an inch above the nominal (base) radius.

The die contours were measured merely in order to obtain reference diameters to determine coating thicknesses rather than to establish absolute contours. The plots developed, however, are accurate representations of the configuration of the surface of the bore. As such, they provide a very important addition: information. Specifically, they provide a record of the distortion or change of each die with time, and given an accurate comparison of die materials, of the number of runs permissible before oversize compacts are produced, and the points at which given die designs and die materials tend to distort first. They also provide information on the probable mechanism by which die dimensions change, e.g., by distortion, by an actual thinning of the die section with attendant metal flow, or by metal removal through wear.

The experience obtained with this method permits making the following generalizations: The thickness of coatings may be determined to an accuracy of about ± .00005" over the range .0001" to .001" thick. The reproducibility of readings is within .00005". The manual methods employed in applying the lubricant result in coatings of consistent thickness, within about .00005" to .0001" depending on the thickness. Each operator applies even coatings of thickness representative of the operator. Coatings vary between operators, some applying much heavier coatings than others.

One of the most limiting characteristics of this method of bore measurement is the large number of careful measurements required. A fertile field for improvement and extension would lie in developing automatic measuring and recording equipment. Available equipment can be adapted for this purpose. There is some limitation in that only dies at room temperature may be measured. The development of a cooled gauge mandrel might permit the inspection of production dies on a routine basis.

BIBLIOGRAPHY

[1]H. M. McCullough & I. Sheinhartz: A Method of Measuring the Film Thickness of Solid Lubricants. (To be published.)

Coming Events

DECEMBER

27-30 American Association for the Advancement of Science, University of California, Berkeley, Calif.

JANUARY, 1955

10-14 SAE Golden Anniversary (annual meeting), Sheraton-Cadillac and Statler Hotels, Detroit, Mich.

24-27 Plant Maintenance & Engineering Show, International Amphitheatre, Chicago, Ill.

FEBRUARY

13-17 American Institute of Mining & Metallurgical Engineers (annual meeting), Conrad Hilton Hotel, Chicago, Ill.

13-18 ASTM Committee D-2 on Petroleum Products & Lubricants, Rice Hotel, Houston, Tex.

MARCH

14-15 Steel Founders' Society of America (annual meeting), Drake Hotel, Chicago, Ill.

APRIL

13-15 American Society of Lubrication Engineers (10th Annual Meeting & Exhibit), Hotel Sherman, Chicago, Ill.

JUNE

26 to July 1 American Society for Testing Materials (annual meeting), Chalfonte-Haddon Hall, Atlantic City, N. J.

Current Literature

(Copies of the following current literature can be obtained without charge by writing direct to the company.)

Industrial Oils & Fatty Acids, Swift & Co., Industrial Oil Dept. (LE10/6), 1800 165th St., Hammond, Ind., 8 pages.

Catalogs all of Swift's industrial oil and fatty acids products. Included are detailed specifications, shipping information, and suggested usage on products of animal, vegetable, and marine sources including oleic acids, red oils, stearic acids, glycerines, hydrogenated and sulfonated products; tallow, lard oils and stearines, as well as data on such specialized products as defoamers, esters, metal and wool processing oils, and Spermaceti USP.

Magnetic & Fabric Filters, Bulletin 350-C, Barnes Drill Co. (LE10/6), 886 Chestnut St., Rockford, Ill., 16 pages.

Illustrates and describes the new Kleenall combination magnetic and fabric filters, utilizing the practibility and low maintenance requirements of separation by permanent magnetic field and positive cleaning by filtration through preselected fabric medium. Twenty point check list serves as a guide in selecting coolant cleaning equipment. Factual application data indicates the results that can be anticipated by maintaining a high degree of coolant clarity. Units range in size from those suitable for single machine application to complete central systems.

Ballbearing Catalogue, No. 54, Split Ballbearing Corp. (LE10/6), Lebanon, N. H., 20 pages.

Shows complete line of Split ball bearings. Gives load and speed ratings, design data, dimensions, etc. Lines shown are single and double row ballbearing bushings in extra-thin section; fully split heavy duty pillow blocks; fully split, heavy duty ball and roller bearings; thin section instrument bearings in torque tube dimensions for precision applications. Information on the various design features possible by fractured race bearings is also included.

The A.B.C. of Modern Lubrication, Bijur Lubricating Corp. (LE10/6), Rochelle Park, N. J., 4 pages.

Covers the three basic elements of automatic lubrication: lubricators, distribution systems, and Meter-Units. Eight types of automatic lubricators are illustrated, with a basic description of a typical pump and reservoir assembly. Distribution system details are given including information on the variety of tubing, junctions, and flexible hoses. The Meter-Unit, which regulates the individual flow of oil at each bearing surface, is described, with a cutaway view showing the internal construction of filter, metering orifice, and check valve. A Bijur offer to develop, without charge, complete designs for built-in central lubrication systems on O.E.M. machinery concludes the bulletin.

A Cylinder Program for Automation & Heavy Duty Service, Miller Fluid Power Co. (LE10/6), 2040 N. Hawthorne, Melrose Park, Ill., 12 pages.

Highlights "standardization" of critical cylinder mounting dimensions and stroke lengths, interchangeable detachable mountings, immediate delivery of a wide variety of sizes and models from stock, highest quality design and specifications, and shows how all of these advantages can be currently obtained for automation and other heavy duty service. The pamphlet contains drawings and tables that demonstrate the large degree of standardization of critical mounting dimensions between

(Continued on p. 352)

FRICTIONAL PROPERTIES OF SOAPS AT HIGH PRESSURES

by J. R. White*

The resistance to shear of lubricant soap films has been measured under normal pressures ranging from 30,000 psi to 250,000 psi. For sodium soaps at pressures below 70,000 psi, the shear strength (coefficient of friction) becomes less as the chain length of the fatty acid component increases from C_4 to C_{18}. At high pressures, the reverse is true and shear strength increases slightly with increased chain length. In regard to temperature, shear strength is less at 100 C. than at 27 C. Above 100 C., the soap films fail to separate the surfaces and shear strength measurements cannot be made.

The addition of long hydrocarbon-chain acids to lubricating oils has been practiced for many years. Largely as the result of the studies of Bowden, Tabor[1], and others, the principal boundary lubricating properties of these acids have been associated with the presence of soaps formed at lubricated surfaces as reaction products between the surfaces lubricated and added acids. Long chain esters, either as the result of incomplete esterification, or hydrolysis, or by direct ester reaction with the oxidized surfaces of metals, may also form soaps at lubricated surfaces and thus possess lubricating attributes similar to acids. Preformed soaps constitute the thickener for the majority of lubricating greases. Many die and drawing lubricants, gear lubricants, and cutting oils contain soaps. Soap and soap forming materials are incorporated in a large variety of products both to enhance their lubrication properties and to secure other desired properties.

Systems lubricated with oils containing acids, esters, or preformed soaps have received much study. Many varieties of apparatus have been devised to assess lubrication as a composite of the properties of an oil and the soap or soap forming materials it contains. The lubricating properties of mineral oils by themselves also have been studied. The lubricating properties of soaps alone, however, have received relatively little consideration.

The lubrication contributions made by soaps are most evident when the lubricated system is operated in what is regarded as the boundary or thin film region. In such a region, should lubrication be ineffective to the degree that wear occurs, the high spots of adjacent surfaces make contact. In general this results in plastic flow of the softer of the metals in the region of the contacts. The maximum pressure reached at the surfaces to be lubricated is then the effective pressure for plastic flow of the softer metal. These pressures, or effective surface stresses, vary from somewhat less than 30,000 psi for softer bearing metals to over 300,000 psi for hardened-steel gear surfaces. This has been taken to be the pressure range of interest for the study of the lubricating properties of soaps.

The present paper reports measurements of the shear strength properties of the even-carbon-numbered sodium fatty-acid soaps from butyrate (C_4) to stearate (C_{18}) at four pressures in the range of 30,000 to 250,000 psi and at temperatures of 27 C. and 100 C. Potassium palmitate, sodium oleate, and an iron stearate have been measured in specified parts of this range.

Apparatus. The apparatus that has been employed in the measurements of the shear strength of soaps is substantially that used by Clark, Woods, and White[2] in the measurement of the shear strength of solidified oils, modified to permit heating of the solids being measured.

This apparatus departs somewhat in basic design from that used by Bridgman[3] and later by Boyd and Robertson[4]. These investigators placed the solid to be tested between two plane surfaced anvils, applied a heavy load, and then measured the torque necessary to twist one anvil relative to the other. In order to avoid the use of a thrust bearing, two such anvil pairs were stacked one upon the other. The thrust was then transmitted through a duplicate source of measured torque. In this manner the unknown and presumably varying friction that might be present in a thrust bearing was avoided. In the present apparatus the same thrust transmission principle has been retained but has been applied to a single spherical anvil contained between two plane anvils which deform plastically to form mating concave surfaces of known area of contact and known high average pressures without the need for large applied loads.

The apparatus is shown schematically in Fig. 1. A photograph showing the physical arrangement of the apparatus is shown as Fig. 12 of reference 2. A hardened steel ball is fastened in a rotor assembly which can be turned with a wire pulley attached to its circumference. Two plane metal anvil surfaces

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This paper was sponsored by the ASLE Technical Committee on Lubrication Fundamentals, and presented at the ASLE 9th Annual Meeting, Cincinnati, April 5, 1954.

are placed above and below the ball and a load is applied until plastic flow of the anvil surfaces creates spherical depressions in the surfaces. These depressions serve the dual purpose of holding the solid whose shear strength is to be measured and of preserving alignment of the assembly during rotation. The torque required to turn the rotor assembly thus originates entirely within the spherical pockets and represents frictional torque from the solid being sheared. This torque is measured by transmitting the driving force on the pulley through a resistance-wire strain-gauge. Normal pressures may be varied by choice of the anvil metal. At convenient pocket diameters of one millimeter, a range of about 30,000 psi to 250,000 psi may be achieved with normal loads not in excess of 350 lbs. These are average pressures; a pressure gradient, discussed in a following section, exists over the pocket area. To permit varying the temperature, independently adjustable electric heaters surround the upper and lower anvils. Thermocouples are placed in the center of the anvils and within two millimeters of the contact areas. The rotor assembly also contains an independently adjustable heater and a thermocouple. For each measurement, both anvils and the rotor assembly are adjusted to the measurement temperature, which is continuously recorded. To avoid constraining the rotor during rotation, the heater and thermocouple leads are removed from the rotor just prior to the measurement. The temperature change in the rotor assembly during the measurement period (about one minute) is inconsequential. Temperatures are believed precise to about 2 C.

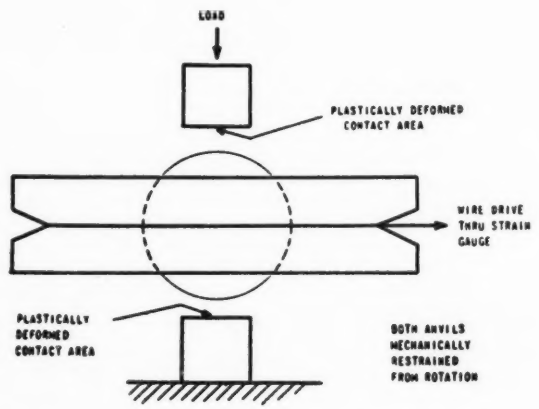

LOAD
PLASTICALLY DEFORMED CONTACT AREA
WIRE DRIVE THRU STRAIN GAUGE
PLASTICALLY DEFORMED CONTACT AREA
BOTH ANVILS MECHANICALLY RESTRAINED FROM ROTATION

Fig. 1

Materials. All soaps have been characterized in terms of the acids from which they were derived.

The sodium soaps were prepared from the previously purified parent acids by titrating carbonate-free sodium hydroxide solution against U.S.P. ethanol-water solution of the acids to the equivalence point. The soaps were then recrystallized from ethanol-water and dried in vacuo at 80 C. Subsequent titration proved the absence of either free acid or caustic.

The iron stearate was obtained by mixing in equivalent proportions ethanol solutions of the acid with an ethanol acetone solution of anhydrous ferric chloride. Upon precipitation, the iron soap was washed with ethanol and dried in vacuo at 80 C. Subsequent titration proved the absence of free acid. Unfortunately hydrolysis during preparation of the iron soap was not completely avoidable. Thus small quantities of basic iron soaps were expected as impurities. The iron soap was an amorphous solid. (Elemental analysis of the iron soap corresponded with a normal soap structure; such analysis, however, is not sufficiently precise to reflect the presence of small quantities of basic soap impurities.)

The fatty acids from which the soaps were made were of E-K "high purity" grade recrystallized several times from U.S.P. ethanol. Equivalent weights determined by electrometic titration coincided with formula weights within 1%. Titration curves revealed only monobasic properties. The oleic acid was of Eimer and Amend C.P. grade similarly analyzed by titration.

Calculation of Shear Strengths. In order to compute the shear strength of a soap, it is necessary to integrate its shear stress contributions over upper and lower contact areas and relate these to the observed frictional torque necessary to turn the rotor assembly of the apparatus. The integration of these contributions must take into account variations in shear strength which may result from the variation in normal pressure over the contact area.

In the absence of an intervening soap layer, the pressures between a sphere and a spherical shell pocket formed by plastic flow of a plane surface may be represented by the ordinates of a hemisphere erected on the area of contact. This has been shown by interference-fringe contour-mapping of the spherical depression after removal of the load. The contours observed were spherical and of larger radius of curvature than the indenting sphere. (Similar results have been reported by earlier investigators using a different method; Ref. 5.) For this geometry Tabor[6] has demonstrated that the Hertz elasticity equations yield elastic deformations approximating those observed. The pressure distribution is then given by the Hertz theory.

In the presence of an intervening soap layer the pressure distribution is less accurately known. If the soap film is thin and comparable with an oil film, contour maps are indistinguishable from those obtained without an intervening film. Very thick soap films, however, may pack irregularly and lead to irregular pressure distribution. Experimentally it was necessary to employ sufficiently thick films so that the mating surfaces were always separated by the film. In general this could be achieved with uniform films of thickness at least greater than that of the surface finish peak to valley irregularities. With such intervening films the hemispherical pressure distribution has been adopted as the best approximation available.

The integration of the shear stress contributions

of the soaps took into account this pressure distribution. An assumption regarding the variance of shear strength with pressure, however, was necessary. Bridgman[3] and Boyd and Robertson[4] have shown that shear strength is an increasing function of normal pressure for essentially all solids. Proportionality between shear strength and pressure is scarcely a precise description of this relationship for all solids, but it is probably the most proper generalization that can be made from available data. In any case the assumption is not highly critical in affecting the shear strengths computed. For example, if the shear strength is assumed independent of pressure, the resulting integrated average shear strength value is only 13 per cent less than that computed if instead shear strength is assumed proportional to pressure.

The shear stress contributions of a soap over the upper and lower contact areas have then been integrated employing a hemispherically distributed pressure, and a shear strength varying in proportion to pressure. The results of the computations have been expressed in terms of ratios of average shear strengths to normal pressures. These dimensionless quantities are convenient for graphing and have a familiar analog in the ordinary friction coefficients. The following equation expresses this relationship (for simplicity, contact areas were calculated as the projected planar areas, an approximation good to a fraction of one per cent):

$$\bar{\tau}/\bar{p} = \bar{f} = 8T/3\pi RW \qquad (1)$$

where τ is the integrated average shear strength; $\bar{p}$ is the average normal pressure; R is the radius of the contact area; W is the applied load; and T is the observed frictional torque for the two contact areas.

The Shear Strength of Soaps. For the quantity $\bar{f}$ in Equation 1 to represent an average shear strength to pressure ratio for the solid being measured, it was necessary to know that the shear in the system was confined to the soap alone and that shear stress contributions from metal surface deformation or slip at the soap-metal interfaces were absent.

To avoid metal surface deformation it was sufficient to interpose soap films of a thickness that prevented sphere to anvil contacts and then to ascertain that such had been avoided by microscopic examination of the sphere and anvil surfaces after each shear strength measurement. Metal contacts were evidenced in circumferential scratching and scoring of the sphere and anvil surfaces. Metal contacts could also be detected by irregularly increasing frictional torques as the sphere was rotated. All measurements exhibiting incomplete sphere and anvil separation were rejected. A typical anvil surface after measurement is shown in Fig. 2. The solid has been partially cleaned from the surface to reveal the microstructure in the spherical depression. Bits of solid not removed in the cleaning are distributed on the perimeter of the depression.

The evidence that the measured torques did not include soap-metal interfacial slip is of necessity less direct. The frictional torque in a given measurement was observed to be reproducibly characteristic of the soap being measured and of the measurement conditions. The torque was independent of the angle through which the sphere had been rotated and within broad limits was independent of the rate of rotation. In addition considerable variation in the metal surface finish effected no change in the observed torque. Microscopic examination made after measurements revealed the soaps as compacted and conforming to the surface finish microstructure of the metal anvils. For surface slip to have occurred would have required simultaneous plastic flow of the soap with stress components that would be expected to vary with the anvil surface finish. Such was not observed. Evidence thus indicates that shear and the associated stress components are confined to the body of the soap.

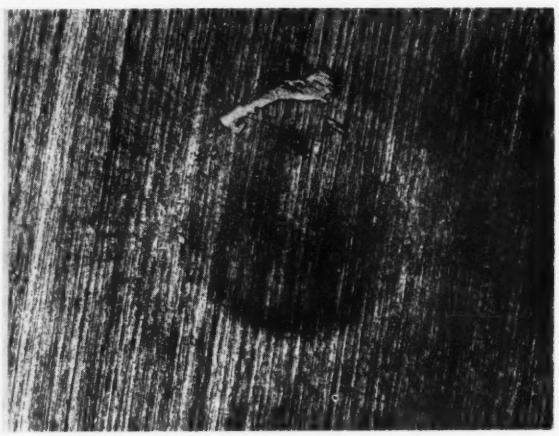


Fig. 2

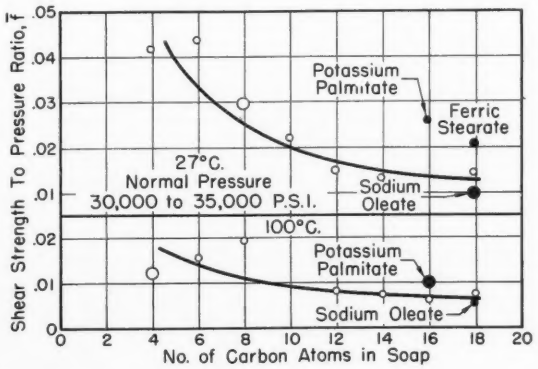
The measurements are presented in a series of graphs. For the sodium fatty acid soaps the ratios, $\bar{f}$, have been plotted against the number of carbon atoms in the soap at each of the measurement conditions. Additional measurements of potassium palmitate, soduim oleate, and the iron stearate are indicated separately on the same graphs as solid circles. Each point plotted is an average of multiple determinations. The radius of the circle circumscribing each point has been made equal to probable error. (Given by

$$0.67 \left\{ \sum (\bar{x} - x)^2/n(n\text{-}1) \right\}^{1/2}$$

where $\bar{x}$ is measurement average, x refers to each measurement, and n is the number of measurements made.)

Graph I depicts measurements at an average normal pressure of 30,000 to 35,000 psi achieved by employing a hardened steel sphere and pure zinc as the anvil metal. Open circles refer to the satur-

ated sodium fatty acid soaps. In general at these lower pressures the shear strengths are observed to increase with decreasing numbers of carbon atoms in the alkyl chain. With pressure held constant, they are slightly lower at 100 C. than at 27 C. The sodium oleate is apparently lower, and the potassium palmitate higher, in shear strength than corresponding saturated sodium fatty acid soaps. Ferric stearate, arbitrarily plotted at an eighteen carbon atom abscissa (three, eighteen carbon atom alkyl chains, are present in this soap), is also higher in shear strength.

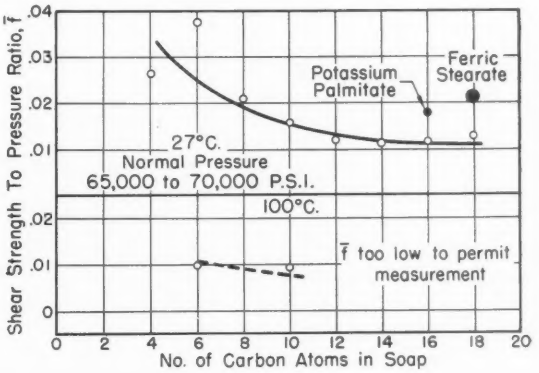


Graph I, curve 384538.

In these and the measurements presented in Graphs II, III, and IV, the solid curves that have been drawn, monotonically relating shear strength to chain length, are arbitrary and intended only to suggest general trends. The mechanical repeatability of each determination is such that deviation of each average from the best smooth monotonic curve that can be drawn must be regarded as real. Choice of alkyl chain length as the single pertinent correlating structural property is thus insufficient to index more than these general trends.

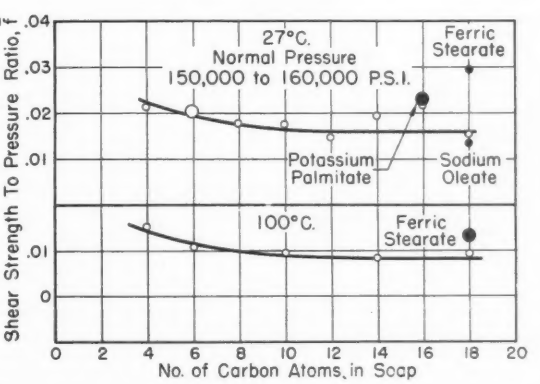
Graph II presents measurements at 65,000 to 70,000 psi obtained when a magnesium alloy constituted the anvil metal. Here, as at the lower pressure, shear strengths trend upward with decreasing alkyl chain length. When temperatures were increased to 100 C., the shear strengths of the higher molecular weight soaps decreased until measurements were not possible; that is, complete surface separation of the ball and spherical shell pocket could not be achieved. (This event occurring only at this pressure among the 100 C. measurements is attributed to metallurgical difficulties with magnesium. Such difficulties are discussed in the next paragraph.)

The inability to achieve surface separation is simultaneously a description of a material limitation and an apparatus limitation. When the shear strength of a solid has decreased until surface separation is marginal, the value at which failure results is of necessity influenced by apparatus variables, such as surface finish, ability of anvil metal to heal after incipient welds, and rotor-anvil alignment, and is thus not entirely characteristic of the material being tested. Certain materials, however, decrease in shear strength so abruptly as temperatures are raised that the shear strength transition resulting in measurement failure cannot be appreciably obscured by apparatus factors. An excellent example is afforded by stearic acid measured at 240,000 psi. At 27 C., 50 C., and 65 C., the $\overline{f}$ values and probable errors are 0.026 ± .0005, 0.020 ± .0003, and 0.018 ± .0006 respectively. At 70 C., however, and all higher temperatures, surface separation cannot be achieved and measurement is accordingly impossible. Stearic acid melts at 69 C.



Graph II, curve 384539.

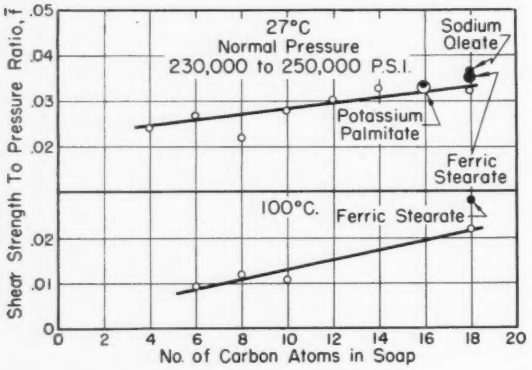
Transition temperatures from lubricated to stick-slip sliding for various acids in oil have been studied by Bowden and co-workers[7] using an apparatus comprised of a constrained slider and a rotating heated table. While the apparatus characteristics of this lubricant testing machine expectedly may enter into the lubricant transition observation, the observed transitions have been sufficiently abrupt to enable assignment of the transition temperature as a lubricant rather than apparatus characteristic.



Graph III, curve 384540.

With perhaps a less well delineated separation of apparatus and material characteristics, the general observation that shear strength decreases with increasing temperature until surfaces cannot be

separated applies to the soaps here studied. Measurements in the range of 150,000 to 160,000 psi and in the range of 230,000 to 250,000 psi have been attempted at a variety of temperatures up to 175 C. In general, while a few successful measurements have been made at temperatures as high as 125 C., generally dependable surface separation has not been achieved at temperatures much in excess of 100 C. Accordingly, the measurements presented in Graphs III and IV do not extend to higher temperatures.



Graph IV, curve 384541.

Graph III contains observations made at a pressure of 150,000 to 160,000 psi using a hardened steel ball and brass as the anvil metal. A definite trend in shear strengths with alkyl chain length is not evident. As before, the iron stearate and potassium palmitate appear more resistant to shear and sodium oleate less resistant than the corresponding saturated sodium fatty acid soaps.

The data in the pressure range of 230,000 to 250,000 psi on Graph IV were obtained with a hardened steel ball and SAE-1020 steel as the anvil metal. At this pressure there appears to be a small but definite trend toward lower shear strength with decreasing alkyl chain length thus reversing the observation at lower normal pressures. Differences between ferric stearate, potassium palmitate, sodium oleate and the corresponding saturated fatty acid soaps are not distinct.

Discussion. In reconciling these shear strength observations with lubrication practice, attention is focussed on the remarkably low values of shear strength that are characteristic of the family of soaps. In addition, these shear strength values all increase with increasing pressure. The higher molecular weight soaps increase in shear strength more rapidly than proportionately to pressure. Attention may also be focussed on the decrease in shear strength the soaps suffer with increasing temperature, thus placing a thermal limitation on their application as lubricants. The observation of increasing shear strength with decreasing alkyl chain length at the lower measurement pressures expectedly enters in the lubrication preference for the longer chain acids and soaps.

A variety of soaps of different metals and acids result from the action of acid-containing lubricating oils at lubricated surfaces. Many soaps thus formed may be expected to possess somewhat different shear strength characteristics than those selected in this study. However, the materials here studied are more or less representative examples of the class of soaps in question, and it is reasonable to expect that their behavior is typical.

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Patent Abstracts

(Prepared from **OFFICIAL GAZETTE** by Ann Burchick. Printed copies of patents are available at 25c each from: Commissioner of Patents, Washington 25, D. C.)

Rust Preventive Compositions, Patent No. 2,687,376, by J. D. Spivack, assignor to Tide Water Associated Oil Co.

A composition comprising a substantially non-corrosive vehicle in major amount based on the weight of the composition and a small amount, sufficient to impart rust-inhibiting properties, of a rust-inhibiting di(N-carboxyalkylalkanamide) in which the alkanoyl radical of the alkanamide grouping is an alkanoyl residue of a long chain fatty acid.

Non-Corrosive Mineral Oil, Patent No. 2,687,984, by E. B. Cyphers & R. H. Jones, assignors to Standard Oil Development Co.

A process for the reduction of bronze corrosivity of highly refined mineral oils containing substantially no corrosive sulfur which comprises treating said oils with from 0.1% to 2.0% by weight of Pb_3O_4 at a temperature within a range of from 250 to 750 F. for a period of time of from 0.25 to 6 hours without removal of any substantial quantity of sulfur from said oil.

Low Temperature Lubricating Composition, Patent No. 2,688,001, by L. S. Echols, assignor to Shell Development Co.

An ash-free low temperature lubricant consisting essentially of a major amount of a mineral lubricating oil and from 1.5 to 3.5% of glycerol monooleate, from 0.3 to 0.5% of lecithin and from 0.2 to 1% of a compound selected from the group consisting of 2,6-ditertiary—butyl-4-methyl phenol, 2,4 — dimethyl — 6 — tertiarybutyl-thiophenol, 2,4,6-tertiarybutyl phenol, 2,2'-methylene bis(4-methyl-6-tertiarybutyl phenol).

Lubricants, Patent No. 2,688,595, by M. Fainman, assignor to Standard Oil Co.

A composition comprising essentially a major proportion of an oil and

(Continued on p. 357)

PHOTOMETRIC EVALUATION OF DETERGENCY OF LUBRICATING OILS

by J. Faust*

A laboratory method has been devised for assessing the detergency of lubricating oils. The results are expressed in terms of a "Detergency Index" and are derived from a photometric determination of the amount of carbon black remaining in suspension after centrifuging a kerosene-diluted sample of oil. The method has been applied to a series of oils containing different types and amounts of petroleum sulfonates, and the ratings were found to correspond to detergency ratings in test engines.

For some time there has been a need for a simple laboratory test to evaluate the detergent properties of lubricating oils. Engine tests used for such evaluation have proved to be laborious, costly, time-consuming and hard to duplicate. Among the more frequently used engine tests are the Caterpillar Diesel Engine Test, requiring 480 hours, and the Lawson Engine tests, requiring either 25 or 44 hours. Despite the variety of laboratory tests hitherto proposed, none has thus far found general acceptance.

In 1939, Bray, Moore and Merrill[1] suggested a simple laboratory method for evaluating detergency in lubricating oils in which specimens of white cloth were soiled in a standard dispersion of lampblack in a lubricating oil containing a detergent additive and the washed cloth examined for grayness. The degree of grayness was taken as indicative of the efficiency of the detergent. This test, however, seems more pertinent to the evaluation of textile scouring agents than to the evaluation of additives for lubricating oils.

Alternative methods for the laboratory evaluation of detergency in lubricating oils have been proposed in more recent investigations. Talley and Larsen[2] proposed use of a chromatographic procedure. By this method, a lampblack suspension of the detergent oil was percolated through a Pyrex tube filled with layers of sand separated by filter paper. The effectiveness of the detergent oil was measured by the number of filter papers darkened by the percolating solution. The accuracy of this method was affected, however, by such variables as the moisture content of the tube packing and the degree of vacuum applied in regulating the penetration of the detergent solution.

Among other procedures suggested by the same authors was the Wood River Test in which a benzene solution of asphaltenes was added to the detergent oil and the mixture put through an asbestos filter. The percentage of (added) asphaltenes which a given oil could carry through the asbestos filter was taken as a measure of the detergency of the oil. The asphaltenes, however, had to be specially prepared by extraction from either used engine oils or oxidized fresh oils and thus varied in nature from one preparation to another.

McNab et al[3] suggested a relatively simple method based on the gravity separation of sludge from the test oil. In this test, the oil was agitated with carbon black and the suspension permitted to settle. The detergent power of the oil was taken as the ratio between the time required for a given amount of carbon black to settle from the oil containing an additive to that required for an equal amount of carbon black to settle from the oil containing no additive. This test was later modified[4] to introduce visual examination and photographic recording of the settling bottles. Discrepancies which appear in this test are the result of deposits which form on the sides of the settling bottles and the inability of the camera to detect material still in suspension. Another disadvantage results from the length of settling time (up to 40 hours) required before the samples may be evaluated.

After extensive investigation of the factors which influence detergency (and the ability of an oil to hold sludge-like products in suspension), the McNab procedure was adopted as the basis of the method herein described. There is, however, one very important difference. Instead of measuring detergency in terms of the amount of oil insolubles settled from the oil, it is measured in terms of oil-insoluble products held in suspension. The amount of suspended material is determined photometrically by measuring the amount of light transmitted through the oil after settling under standardized conditions. The results obtained from it are referred to as a "Detergency Index." An important part of the test is the retention of the use of carbon black instead of lampblack as the oil insoluble material.

DETERGENCY OF LUBRICATING OILS. Before describing the method in detail, it is important to clarify the definition of "detergency" used in this paper. As has been pointed out by Carl W. Georgi,[5,6] the term detergency is a misnomer as applied to crankcase lubricants. What is actually meant is the ability of the oil to suspend dirt or

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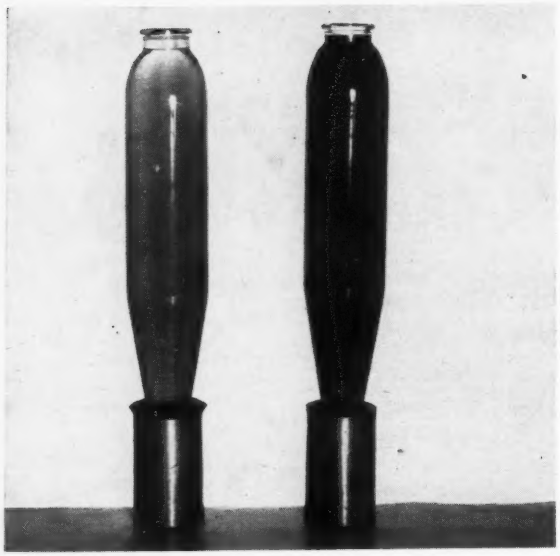

Fig. 1 Centrifuged kerosene-diluted oil samples.

sludge as it may accumulate and maintain these oil insoluble products in suspension. On this basis, the function to be tested could more properly be called "dispersancy," and the present task is to develop a simple laboratory procedure for the evaluation of dispersant action.

Detergency Index Test: The method is based on the following principle:

An oil which has little or no dispersant properties will allow oil insoluble products, such as sludge, soot, dirt, etc., to settle rapidly. When this oil is decanted after settling and tested in a photometer, it will show a high light transmission. On the other hand, an oil which possesses dispersant properties will keep most, if not all, of the insolubles in suspension. It will be clouded and show a low light-transmission. The amount of light transmitted through a sample oil, that has been allowed to settle under standardized conditions, thus becomes a measure of the dispersant power of the oil, and can be evaluated quantitatively by conventional photometric means. In order to avoid errors due to dirt accumulated on the walls of the glass container used for measuring cloudiness, photometric measurements should be made promptly after placing the sample in such containers.

Experimental: A large number of tests were carried out before the final procedure of testing was adopted.

To simulate the condition of oxidized oil carrying sludge and fuel combustion products, a "sludged" or "black" oil is prepared according to a standard procedure. "Molacco" type carbon black, supplied by Binney & Smith, was selected to simulate "sludge-like products." This almost anhydrous type of carbon black was found to be a distinct improvement over the "Excelsior" type, whose high moisture content (6%) was observed to affect the reproducibility of results adversely.

In order to accelerate the settling of the sludge-like products and reduce testing time, it was found useful to dilute the test oil and centrifuge the diluted blend. Precipitation naphtha was originally used as a diluent but it was found later that any water-white petroleum fraction, such as kerosene or low-viscosity white mineral oil, would also serve satisfactorily. The provision must be made, of course, that once the type and amount of a diluent have been chosen, both must be held constant throughout a given series of tests. Water-white kerosene with the following specifications was finally chosen as the diluent for the laboratory procedure:

Specific Gravity	0.811
Flash Point	135° F.
Fire Point	150° F.
Distillation Range	
Initial Point	360° F.
50 cc	440° F.
End Point	530° F.

As to the choice of a photometer, any of the commercially available photometers that will measure directly the percentage of light transmitted through an oil may be used. The photometer used in most of the present experiments was a homemade instrument, equipped with a constant light source (lamp bulb enclosed in a well-ventilated housing), a photocell of the self-generating type and a low-resistance microammeter. The scale on the microammeter is divided into 100 equal divisions which are arbitrarily accepted as corresponding to transmittancy readings ranging from 0 to 100%. No attempt was made to check the linearity of the relationship between the intensity of light and the microammeter reading. A rectangular glass cell, 50 mm long, 10 mm wide and 50 mm deep, as supplied with the Fisher Electro-photometer, serves as the receptacle for the oil samples (see photo). These glass cells are held firmly in a close-fitting compartment to assure proper position with reference to the light and the measuring photocell. The light intensity of the lamp is adjustable by a rheostat to give a reading of 100 with the cell filled with the kerosene diluent plus oil. The instrument is therefore operated at all times at the same sensitivity level.

The centrifuge tubes used in centrifuging the oil samples are the same as those used for the Pre-

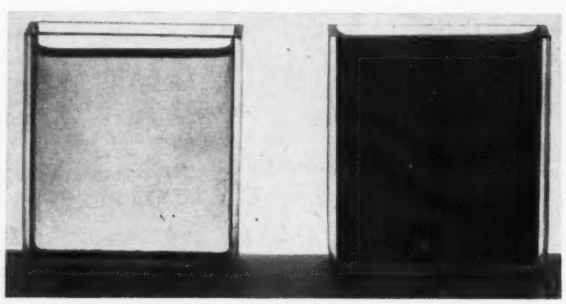

Fig. 2 Photometer cells containing centrifuged kerosene-diluted oil samples.

cipitation Number determination of lubricating oil under ASTM D-96-40. See Fig. 1 for illustration of centrifuged samples.

Test Procedure: 50 cc of the oil to be tested are diluted with 50 cc kerosene and agitated with 0.5 grams of carbon black in a 250 ml beaker for exactly five minutes. Agitation should be effected by a small-sized, speed-controlled agitator at a speed of 1,100 rpm. This "sludge oil" is immediately diluted with kerosene in a ratio of 25 cc sludge oil to 75 cc diluent and shaken in a glass-stoppered 100 cc glass cylinder with 20 complete up-and-down strokes. The diluted "sludge oil" is then poured into a centrifuge tube up to the 100 cc mark, and the tube centrifuged at approximately 1,700 rpm for one hour. The centrifuge used (from the International Instrument Company) has a radius of 5¼ inches from centre to pivot point.

After centrifuging, the oil is immediately decanted into the rectangular glass cell, see Fig. 2, and put into the photometer for light-transmission determination. The photometer must be adjusted prior to each determination to give a reading of 100 microamperes for light transmitted through the blank, which is the kerosene diluted test oil minus carbon black. A reading (in microamperes) is then taken for the test oil, and this figure subtracted from 100, the difference being recorded as the Detergency Index. To illustrate:

	Light Transmission in Microamperes
Kerosene + test oil (but no added carbon black) (Blank)	100
Oil sample under test	37
Difference in reading (Detergency Index)	63

The difference between the first and second reading (in this case 63) is due to the light "blocked out" by the sludge-like particles maintained in suspension in the oil. The value 63 is referred to as the Detergency Index of the oil sample under test.

As an example of some actual test figures, results are listed below which were obtained in testing a conventional Pennsylvania Oil of SAE 20 containing a commercially available detergent additive and the same oil without the additive.

Table I

	Detergency Index
Oil without detergent	25
Oil with detergent	63

It is to be noted that the Detergency Index is high for an oil of high dispersant power and low for an oil of little or no dispersant properties.

Investigations included an attempt to determine the correlation between Detergency Index values and results obtained in actual engine tests. For this purpose, three detergent-type lubricating oil samples were secured which had been tested for degree of detergency by several laboratories under Specification MIL-0-2104. Compartive data for these tests are given in Table II.

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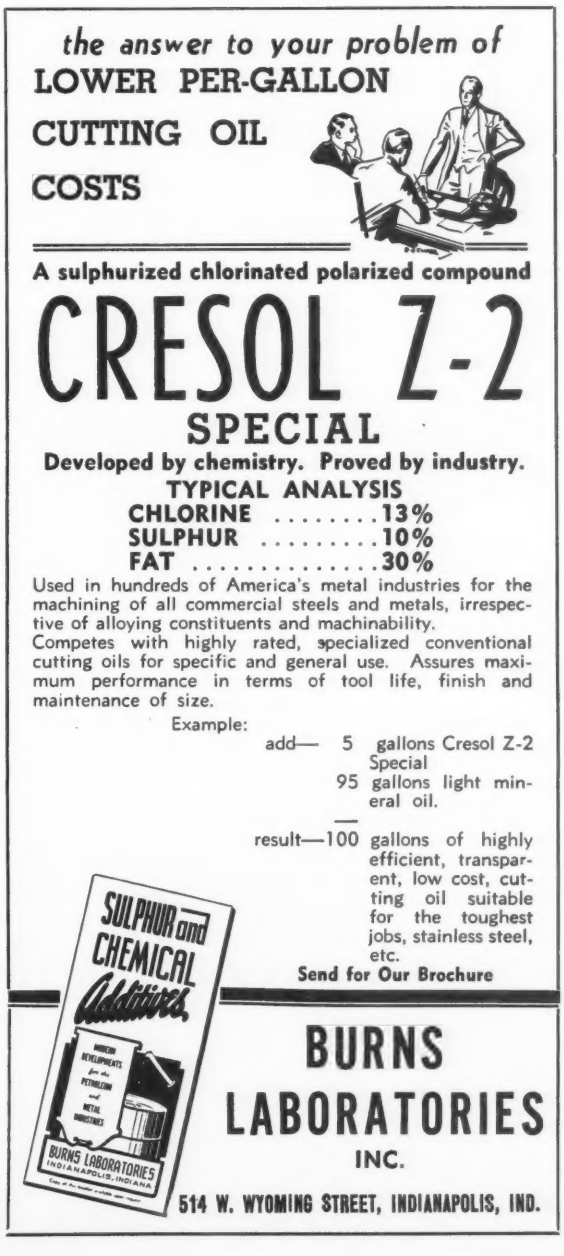
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Table II

	Oil Sample		
	No. 1	No. 2	No. 3
Detergency Classification by Engine Test	poor	intermediate	good
Detergency Index Test	31	47	76

Detergency Index values are here seen to parallel results obtained in actual engine classification tests.

Rating of Detergent Additives: Detergent Index values may be effectively used in rating detergent additives during laboratory development work of these materials. Such relative rating is illustrated by data given in Table III.

Table III

	Detergency Index
Oil (a Pennsylvania SAE 20 Motor Oil)	30
Same oil plus:	
0.28% Sodium Sulfonates of 430 mol. wt., Sulfate Ash 16.6% (a)	28
0.28% Sodium Sulfonates of 450 mol. wt., Sulfate Ash 15.8% (a)	28
0.28% Calcium Sulfonates, Sulfate Ash 9.5% (b)	70
0.28% Barium Sulfonates, Sulfate Ash, 20.5% (b)	45
0.28% Commercial Additive No. 1 (c)	71

Key: (a) All sodium sulfonates were used in form of a 62% solution of petroleum sulfonates in oil.
(b) The calcium and barium sulfonates were made from (a) and used in form of a 50% solution oil.
(c) A calcium salt of a petroleum sulfonate.

These data indicate the possibilities and usefulness of the Detergent Index as an aid in screening out nondispersant materials in preliminary laboratory development work on detergent additives. It would also appear from these (and other) data that lubricating oils containing commercially available detergent additives give Detergent Index falling between 65 and 100, while lubricating oils without additives, having poor or no dispersant properties, give Detergent Index values of below 65.

Concentration of Additives: Detergency Index values also indicate the effect of varying concentrations of an additive on dispersive power.

Table IV

	Detergency Index
Oil (a P a, SAE 20 Motor Oil)	30
Same oil plus:	
0.28% Sodium Sulfonates of 450 mol. wt.	28
0.56% Sodium Sulfonates	28
1.12% Sodium Sulfonates	28
0.28% Calcium Sulfonates	70
0.56% Calcium Sulfonates	77
0.80% Calcium Sulfonates	85
1.20% Calcium Sulfonates	96
0.28% Barium Sulfonates	45
0.56% Barium Sulfonates	55
0.80% Barium Sulfonates	85
1.20% Barium Sulfonates	96
0.28% Commercial Additive No. 1	71

The effect of higher concentrations on the dispersive power of additives tested is clearly indicated by their respective Detergency Index ratings. It is best, however, to test detergent additives in low

concentrations if differences in "detergent response" are to be brought out significantly.

Reproducibility of Results: With proper care and attention to details, duplicate determinations of the Detergency Index of lubricating oils should not differ by more than 5 microamperes readings. This is illustrated by data given in Table V.

Table V

	Detergency Index (a)	(b)
Oil without detergent	29	32
Oil plus additive No. 1	70	66
Oil plus additive No. 2	58	62
Oil plus additive No. 3	68	63

SUMMARY. The Detergency Index is a numerical value expressing the ability of a centrifuged, kerosene-diluted, detergent-type lubricating oil to hold carbon black in suspension. A high Detergency Index rating is indicative of high-sludge dispersant power, while a low rating indicates that the oil has little, if any, power to keep oil insoluble products in suspension. To the extent that the method has been tested in this laboratory, it shows good correlation with engine tests of detergency and rates additives in line with general experience. The Detergency Index has been used in this laboratory for a number of years and has been found satisfactory for such tasks as differentiation between detergent and nondetergent oils, evaluation of the dispersant power of commonly used commercial lubricating oil additives, and investigation of proposed additives.

The procedure is simple, rapid, requiring not more than 1½ hours and may be carried out with very little special equipment. The reproducibility of results obtained under standardized conditions of this test is good, measurements varying within ± 5 microamperes readings.

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(Section News, from p. 318)

by Dr. H. E. Mahncke, Westinghouse Research Laboratories; "Lubricant Tests," by C. E. Bailey, National Tube Div., U. S. Steel Corp.; "Bearings—Gears," by W. T. Everitt, Eastman Kodak Co.; "Organization of Lubrication," by J. D. Lykins, Wheeling Steel Corp.; and a Panel Discussion on Lubrication problems. (Submitted by W. R. Johnston, Sec'y.)

KINGSPORT, March. Election of Officers (See ASLE Directory).

September. E. M. Higgins, Master Lubricants Co., presented a paper entitled "Anti-friction Bearings Operations." (Submitted by J. E. Fleenor, Sec'y-Treas.)

MILWAUKEE, September. T. P. Sands, Monsanto Chemical Co., presented a paper entitled "The Development of Automotive Gear Lubricants—Present & Future," in which he discussed current test methods for the performance evaluation of automotive gear lubricants, the performance of present-day base oil-additive combinations in controlled tests and in the field, and the probable future gear lubricant requirements related to the current development of superior oil-additive combinations.

October. A roundtable discussion on "Soluble Oils," with W. E. Elliott, Allis Chalmers Mfg. Co., acting as monitor. (Submitted by J. A. Gramling, Sec'y.)

PHILADELPHIA, May. Election of Officers (See ASLE Directory. Submitted by R. W. Clark, Sec'y.)

TWIN CITIES, September. S. D. Craine, W. A. Jones Foundry & Machine Co., presented a paper entitled "Gears and the Lubrication Engineer."

October. Dr. T. W. Langer, Union Carbide & Carbon Corp., spoke on the subject "Synthetic Lubricants in Industry," followed by R. R. Burford & V. E. Welschinger, Minnesota Mining & Mfg. Co., who presented a paper entitled "Fluorochemical Fluids for

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Sight Feed Lubricators." (Submitted by C. D. Johnston, Sec'y.)

YOUNGSTOWN, September. B. M. Dunham, Sun Oil Co., presented a paper on "Compressor Lubrication" in which he classified compressors currently used in industry according to the approximate range of speeds, pressure developed, and air capacity, together with the lubrication and approximate viscosity of lubricating oils which have proved most suitable.

October. A plant tour, conducted by C. Morgan, through the Copperweld Steel Corp., of Warren, Ohio. (Submitted by W. T. Williams, Sec'y.)

(Current Literature, from p. 339)

various cylinder manufacturers, and a table that gives internal fluid pressure stresses of essential cylinder components.

Labline-Seta Petroleum Testing Apparatus, Labline Inc. (LE10/6), 217 N. Desplaines St., Chicago 6, Ill., 48 pages.

Catalogs petroleum testing apparatus made by Stanhope Engineering Co. of London, England, including flash testers, Penetrometer, gum apparatus, grease testing apparatus, and specials. (Labline Inc. is the U.S.A. distributor of Stanhope products, marketed under the trade-name of Labline-Seta.)

(Lubrication Summarized, from p. 326)

Tricresyl phosphate, known also as Lindol HF. Special care must be taken in selecting packing and gasket stocks. Normal applications are adequately taken care of by butyl rubber base synthetic rubbers. Silicone base compounds have been used with great success in some cases. But softening or tenderizing can be expected on both compounds at temperatures above 212 F.

Carbonated diphenyls, of which Arochlor 1248 is typical. At regular temperatures, Buna N, GR-S butyl rubber, or silicone base compounds are satisfactory. But the field narrows to silicone base compounds as temperatures are increased. Careful selection of the compound to the application is stressed again here.

Soluble oils, produced by various oil companies. None of these present a particular problem to the rubber compounder. The correct compound can be supplied for each application. Buna N's, neoprenes, and silicones can be used and can be properly sealed through a temperature range of —65 F. to 350 F. with the organic polymers. The temperature range can be enlarged with use of Teflon and Kel-F.

(New Products, from p. 335)

Corp., Instrument Div. (LE10/6), 1828 W. Diversey Blvd., Chicago 14, Ill.

POROUS TEFLON FILTERS. By using a unique sintering and molding process without the aid of any additives or plasticizers, porous Teflon filter medium is being produced which has all the outstanding corrosion resistance and temperature stability properties of solid Teflon. Porous Teflon has a mean pore size of 9 microns and will filter out of liquids all particles having diameters larger than 3 microns. In gas filtration all particles as small as one micron will be removed. Flow capacities as high as 45 gallons per minute per square foot of filter at 5 psi differential pressure can be anticipated for organic fluids of 1 cp viscosity. This new material is available in disc form in sizes up to 12 inches in diameter with a solid Teflon rim for high tear strengths and providing for integral gasketed mounting. For further details, write: Porous Plastic Filter Co. (LE10/6), 31 Sea Cliff Ave., Glen Cover, L. I., N. Y.

VANTROL 5618 CLEANER. A new, low-priced, highly effective spray-type cleaner that combines solvent, wetting agent, and alkaline as a multi-phase cleaner has been announced that is especially effective on steel. In addition

At ZERO temperature here's proof that

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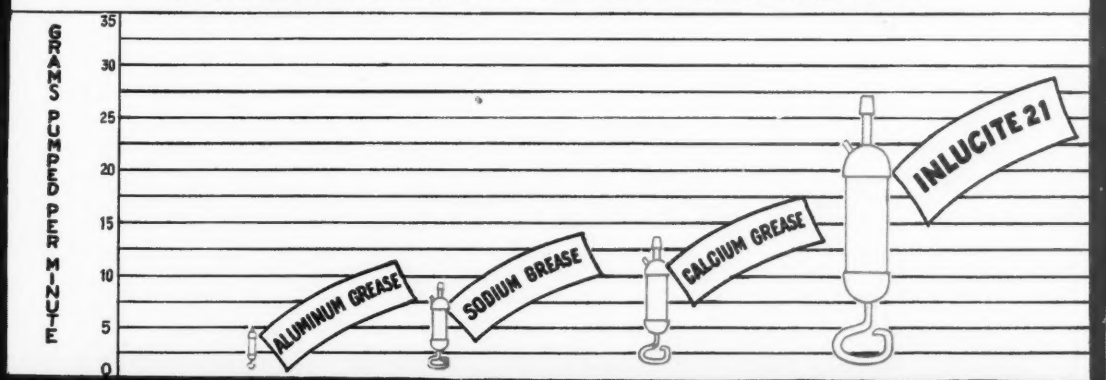
pumps freely when other greases freeze

PUMPING RATE AT 0° F. OF TYPICAL CHASSIS GREASES IN STANDARD AUTOMOTIVE GREASE DISPENSING EQUIPMENT



Typical chassis greases were placed in a conventional service station type grease gun, and left for two weeks in a cold room at 0°F. The ASTM No. 1 grade of aluminum-base chassis grease hardened so that it was almost impossible to pump.

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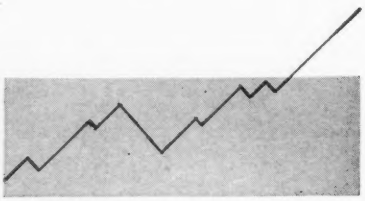

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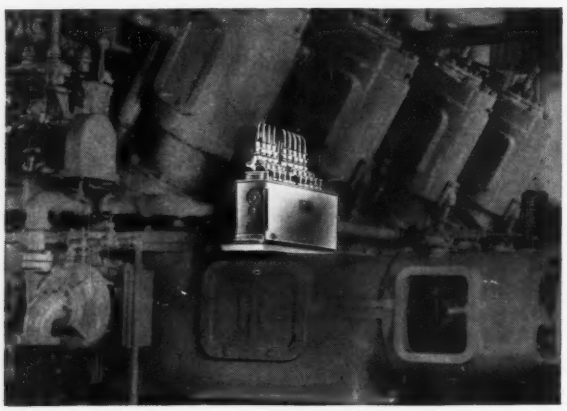

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to its effective removal of shop and mill oils, drawing compounds, and heavy greases, the new industrial cleaner has the added value of rust protection. It is normally used at from a half to two ounces per gallon. When used at a quarter-ounce per gallon in the rinse, it gives steel adequate rust protection. For complete details, write: VanStraaten Chemical Co. (LE10/6), 82 W. Washington St., Chicago 2, Ill.

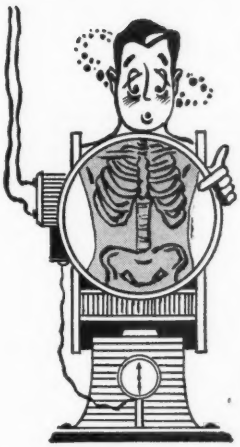
BIG 'D' DOPE. A single application of this newly developed lubricant-seal provides permanent lead-plating, island high, permitting repeated quick assembly and disassembly of any size threaded connections without re-doping and without damage to threads. Guaranteed to afford savings in excess of 70% on materials and man-power, Big 'D' has passed the most rigid tests as a lubricant-seal for aircraft and marine equipment, hydraulic lines, drill pipe and tool joints, nuts and studs on all types of pumps and engines, large plug valves, fork-lift and elevator guide channels, heavy open-worm gears, line grooves on sheaves, and as a packing sealer and steel-cable lubricant. The compound is warranted anti-gall, non-corrosive, non-hardening and leak-proof, and resistant to steam, gas, oil, air, chemicals, alkali, acids, ammonia, and all hydrocarbons under the most punishing pressures, temperatures, and vacuums. For complete details, write: C. H. Dragert Co. (LE10/6), Dallas, Tex.

MO-SILICONE VACUUM GREASE. A new principle in lubrication of vacuum apparatus has been developed which combines the chemical inertness and temperature-viscosity relationship of silicone liquids with the lubricating qualities of molybdenum disulfide. The silicone, a relatively thin liquid, is thickened to a grease-like consistency by the addition of Lockrey-processed submicronized MoS_2, so that it serves as a tight seal against both high and low pressures, as well as a highly efficient lubricant for moving parts. Air-tight, easily as-

sembled glass-to-rubber joints can be made with this material. It also serves as a seal and thread compound for making tight threaded joints in stainless steel, glass or other chemical pipe-work, as well as making such joints easy to disassemble without damage when changes are necessary. For further details, write for Bulletin 25-A, The Lockrey Co. (LE 10/6), Southampton, N. Y.

MICRO-FOG LUBRICATORS. A complete new line of Micro-Fog lubricators for air tools, air cylinders, and other air-operated equipment has been announced featuring an easily adjusted variable vane venturi which enables one unit to meet the requirements of a much wider range of operating conditions, and handle applications which previously required several sizes of lubricators. For example, at 80 psi the ⅜" size can be set for a low flow range of 5 to 9 cfm, a high range of 50 to 100 cfm, or any intermediate range desired. The new lines includes 20 models in pipe sizes of ¼", ⅜", ½", ¾", and 1", with capacities of ½ pt., 1¾ gal., and 4½ gal., and flow range from 5 to 250 cfm at 80 psi. For complete details, write for Bulletin 496, C. A. Norgren Co. (LE10/6), 3434 S. Elati St., Englewood, Colo.

LUBE MEASURING VALVES. Hydraulically-operated valves for dispensing predetermined, measured quantities of lubricant are now available with any discharge output range desired. Designed for use with a four-way control valve, which is supplied with lubricant from a power-operated pump, these valves incorporate an adjusting screw at each end of the measuring chamber for controlling the exact quantity of lubricant output desired, and mounting brackets to facilitate installation. The system provides means for automatically applying metered quantities of oil or grease to gear cases, small transmissions, bearings, electric motors, etc., in assembly line operations. For specifications and prices, write: Lincoln Engineer-



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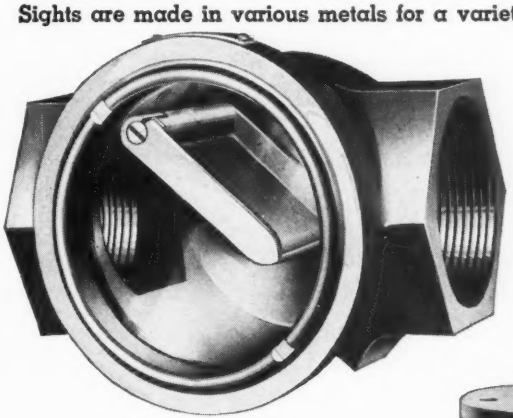

FIG. 811

This vane-type Bowser Indicator is spring-actuated, thus providing an indication of the quantity flowing in the line.

FIG. 811

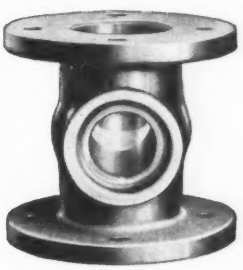

FIG. 54A

This Bowser Flow Sight is recommended for use on gravity lines where the flow is vertically downward.

FIG. 54A

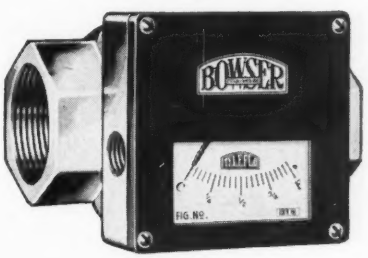

FIG. 816

The Figure 816 Bowser Teleflo Indicator is equipped to operate a gong or to stop pump motor if liquid flow stops.

Consult with the Bowser engineer nearest you.

FIG. 816

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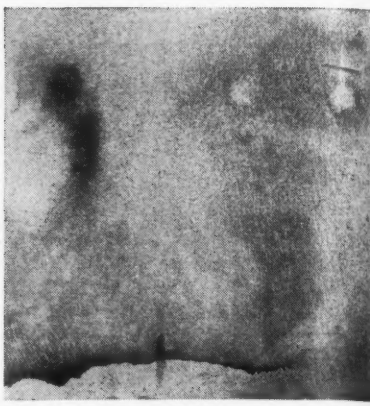

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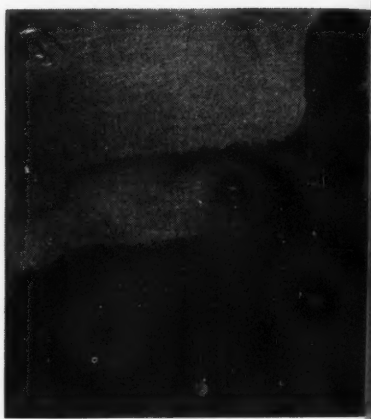

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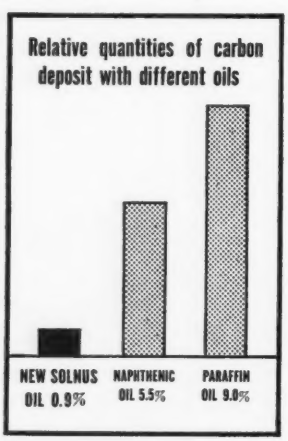

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(Patent Abstracts, from p. 344)

an oil-soluble aromatic amino salt obtained by reacting an arylol carboxylic acid ether with an aromatic amino compound selected from the class consisting of a naphthylamine, a phenylenediamine and pp'diamino diphenyl methane, said salt being used in small but sufficient amounts to inhibit the oxidative deterioration of said oil and to impart a corrosion-inhibiting property to said oil.

Lubricating Oil Additives, Patent No. 2,688,596, by E. M. Deffley, assignor to Standard Oil Development Co.

A process for the stabilization of polymers of vinyl ethers containing from 3 to 12 saturated carbon atoms which comprises heating a material consisting essentially of an oil solution of said polymer to a temperature within a range from 50 to 100 C. for from 20 to 60 hours, said polymer having a molecular weight within the range of from 20,000 to 35,000.

Process for Preparing Insulating Oil Composition, Patent No. 2,688,597, by A. F. D. Pott & D. H. McLean, assignors to Standard Oil Development Co.

The process of preparing insulating oils of low power factor and high insulating properties which consists essentially of first treating the oil with 0.05 to 2% of 2,6,di-tertiary butyl, 4-methyl phenol and thereafter contacting said treated oil with a finely divided solid adsorption agent and filtering, said contacting being accomplished at a temperature between 50 and 250 C.

Preparation of Phosphorus Sulfide- Hydrocarbon Reaction Products and Removal of Inorganic Phosphorous Acids Therefrom, Patent No. 2,688,612, by R. W. Watson, assignor to Standard Oil Co.

In the preparation of an oil-soluble lubricant addition agent wherein a normally liquid hydrocarbon is reacted with about 1% to about 50% of a phosphorus sulfide at a temperature of from about 200 F. to about 600 F. and the resultant reaction product hydrolyzed at a temperature of from about 220 F. to about 500 F. whereby inorganic acids of phosphorus are formed, the improvement comprising contacting said hydrolyzed reaction product with an alkaline adsorbent material selected from the group consisting of fuller's earth, diatomaceous earth, bentonite, magnesite and bauxite, at a temperature of from about 100 F. to about 500 F. for a time sufficient to remove the inorganic acids of phosphorus formed by said hydrolysis, and separating said contacted liquid from said adsorbent, whereby an oil-soluble reaction product of a phosphorus sulfide and a normally liquid hydrocrabon, substantially free of inorganic acids of phosphorus, is obtained.

Soluble Cutting Oil, Patent No. 2,689,-222, by J. D. Oathout, assignor to Standard Oil Development Co.

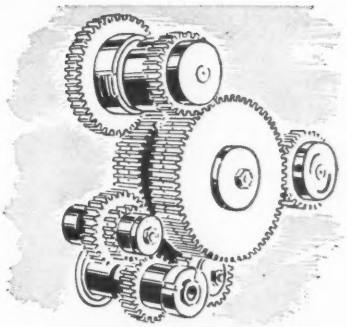
A soluble metal working oil composition comprising a major proportion of mineral base lubricating oil, a minor emulsifying proportion of an oil-soluble sulfonate, a minor proportion of sulfurized fatty oil of 6-20 wt. % sulfur content, sufficient to increase the load-carrying capacity of said composition, the weight ratio of said sulfonate to said sulfurized fatty oil being about 1:2 to about 2:1, and about 0.1 to about 3% by weight of total composition of a polyoxyethylene fatty acid tri-ester of sorbitan.

Viscosity Index Improvers, Patent No. 2,689,223, by A. H. Popkin & T. S. Tutwiler, assignors to Standard Oil Development Co.

A lubricating oil concentrate useful for blending purposes which consists essentially of a mineral base lubricating oil containing combined therein from 10% to 90% by weight of a copolymer of a low molecular weight aliphatic isoolefin and a low molecular weight aliphatic conjugated diolefin, said copolymer having viscosity index improving characteristics and having been reduced in molecular weight from one within a range of from about 40,000 to 250,000 Staudinger to one within a range of from about 10,000 to 30,000 Staudinger and containing a minor amount, sufficient to improve the thermal stability characteristics thereof, of a phosphorus sulfide reaction product of a metal salt of an alkyl phenol sulfide.

Friction Reducing Additives for Lubricants, Patent No. 2,689,224, by A. F. Sirianni & I. E. Puddington, assignors to The Honorary Advisory Council for Scientific and Industrial Research.

A lubricating composition for heavy journal metallic bearings operating at elevated temperatures in the presence of moisture, comprising a mineral oil base lubricant containing 0.1 to about 1% by weight, based on the total composition, of a fatty acid modified resin selected from the group consisting of the condensation products of glycerol with a dicarboxylic acid material selected from the group consisting of phthalic, succinic and maleic acids and their anhydrides modified by treatment with long chain fatty acids having about 12-22 carbon atoms, and the condensation products of C_{12}-C_{22} long chain fatty acid esters of glycerol with said dicarboxylic acid materials, said resin having free acidity to the extent of an acid number of about 7-100 and having sufficiently active carboxylic groups to form substantially oil-insoluble chemi-sorption products with the metal of said bearings at the operating temperature, said resin being substantially devoid of free fatty acid which would form oil-soluble soap with said metal.



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No. 155

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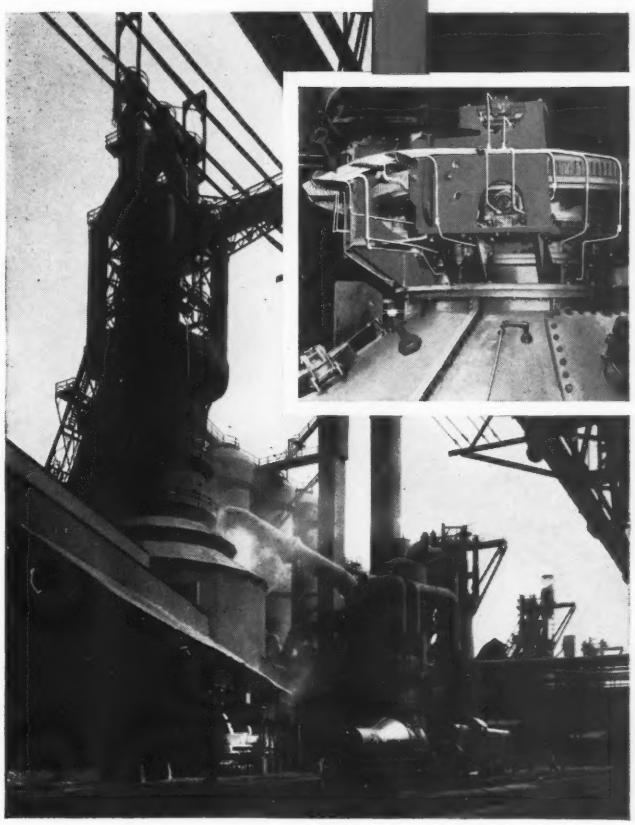

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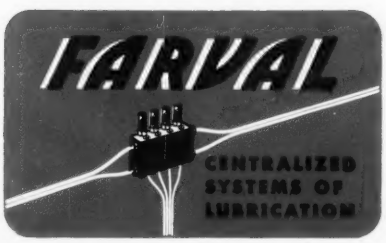

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